BUFFALO FAN SYSTEM

HEATING
VENTILATING
HUMIDIFYING
DRYING

CATALOG 198

BUFFALO FORGE COMPANY

BUFFALO, N.Y.

CATALOG NO. 198

BUFFALO FAN SYSTEM

OF

HEATING AND VENTILATING

PART I. HEATING AND VENTILATING OF PUBLIC BUILDINGS.

PART II. HEATING AND VENTILATING OF INDUSTRIAL BUILDINGS.

PART III. BUFFALO HEATING AND VENTILATING APPARATUS.

PART IV. DATA ON HEATING AND VENTILATING.

BUFFALO FORGE COMPANY

ENGINEERS AND MANUFACTURERS

BUFFALO, N. Y., U. S. A.

WORKS

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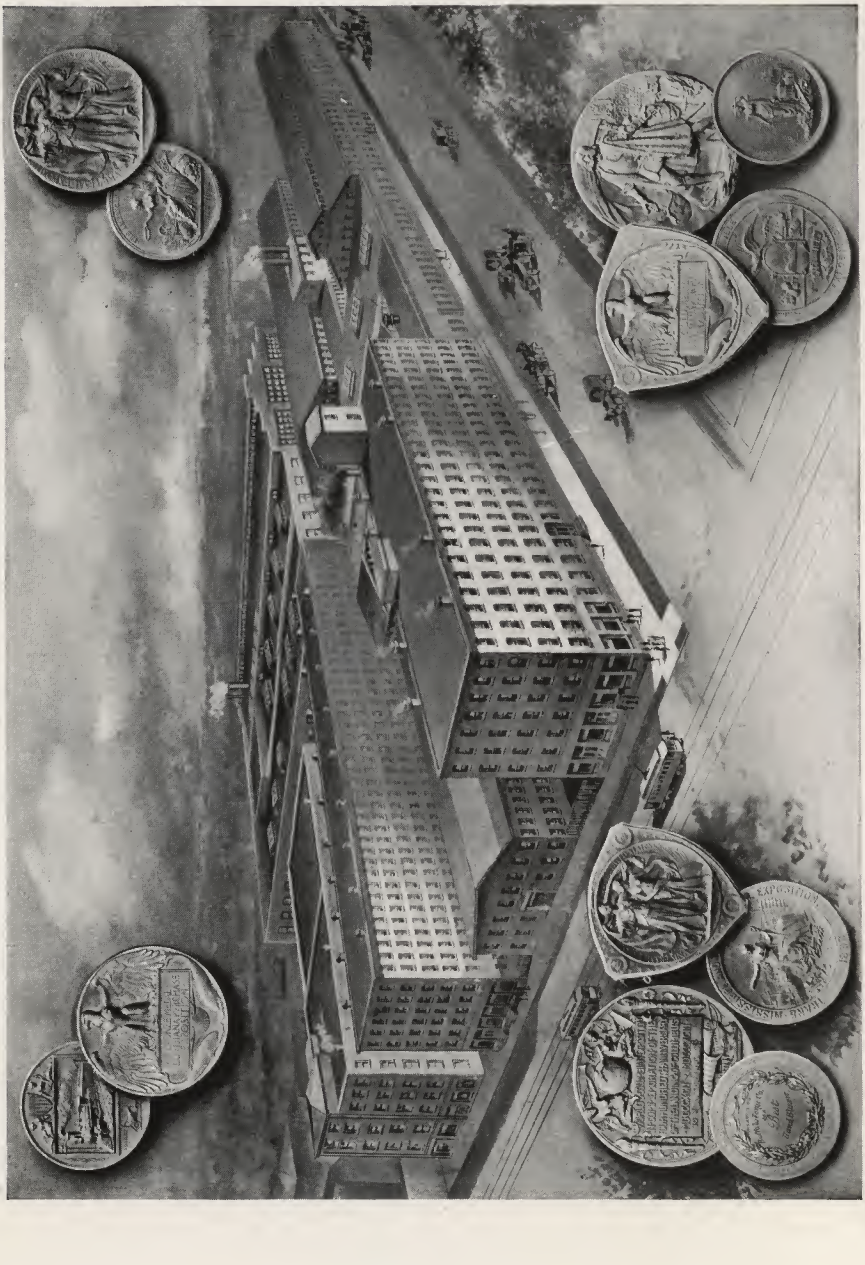


MANUFACTURING PLANT OF THE BUFFALO FORGE COMPANY, BUFFALO, NEW YORK, U. S. A.

PREFACE

Arrangement

Architecture and engineering in this twentieth century are more closely allied than ever before. The engineering profession, emerging from the utilitarian, finds its services required in all branches of the so-called fine arts and its ingenuity taxed for the development of details heretofore overlooked or slighted. It is fortunate that the distinct advance and wider application of mechanical heating, ventilating and air purifying methods have, under the influence of our foremost architects, physicians, and others interested in the public health, kept step with the increasing need. That the need exists, there can be no doubt. Scientific analysis is not required to show the presence of impure air, dust and smoke. It is therefore an exception to find a public building, school or factory which has not suitable provision for mechanical ventilation, whether in connection with or in adjunct to the heating system. To the engineer and to the architect, who for our present purpose are practically the same, as well as to the factory owner and to the layman interested in this subject, we offer this book, combining the catalogue and the treatise.

Sub-Divisions

Attention is called to the title page, showing that for convenience of reference, four sub-divisions have been made. Part I deals with heating and ventilating public buildings. Part II with industrial buildings, which offer a quite different problem. Part III is devoted to the apparatus itself and Part IV to general data on its application to meet special requirements. No effort has been spared to make the descriptions full and clear for the benefit of readers unfamiliar with the system and the insertion of engineering information published only by this company, will make this book of service to engineers, architects, and other technical readers; we refer particularly to data on transmission of heat and to the revised data on friction of air and equalization of piping systems which are practically the only published results of actual tests in the United States.

The facts and figures found in these pages are from records—carefully compiled and verified from actual tests secured at our testing laboratory and in numerous plants throughout the United States from 1901 to 1914, by our engineering and testing department. Detailed data covering special conditions will be furnished on application.

Important Features

Besides the data made available there are some other important features which should prove of the greatest interest.

1st. The Carrier Air Washer and Humidifier, manufactured by the Carrier Air Conditioning Company of America, 39 Cortlandt St., New York City. Carrier Air Washers are now being used extensively in connection with Buffalo Fans and Heating apparatus for air purifying and humidifying, in state and national public buildings, schools, theaters, hotels, churches, office buildings, etc. This system has met with great success for humidifying work in cotton, worsted and silk mills, paper factories, tobacco plants, malt houses, bakeries, and is very adaptable for keeping proper amount of moisture in mines, preventing dangerous explosions and fires. Carrier Dehumidifiers have answered a long felt want in film drying plants, macaroni plants, capsule drying plants, candy factories and printing establishments.

2nd. Buffalo Niagara Conoidal Type "N" Fans and Turbo Conoidal Type "T" Fans which are fully described on page 66. These fans are a distinct advance in design, and show much better efficiencies than any other type of multiblade fan. This line is the direct result of extensive tests made under hardest conditions.

3rd. Buffalo Steel Plate Planoidal Type "L" Fans, an outgrowth of Buffalo Steel Plate Fans, which show high efficiency, which is obtained by the use of carefully designed cone inlets, and proper proportioning of parts.

4th. The Buffalo Gas Heater is explained on pages 83 to 85 and represents an entirely new field for the application of the fan system where natural gas can be obtained, producer gas used or where manufactured gas is moderate in price.

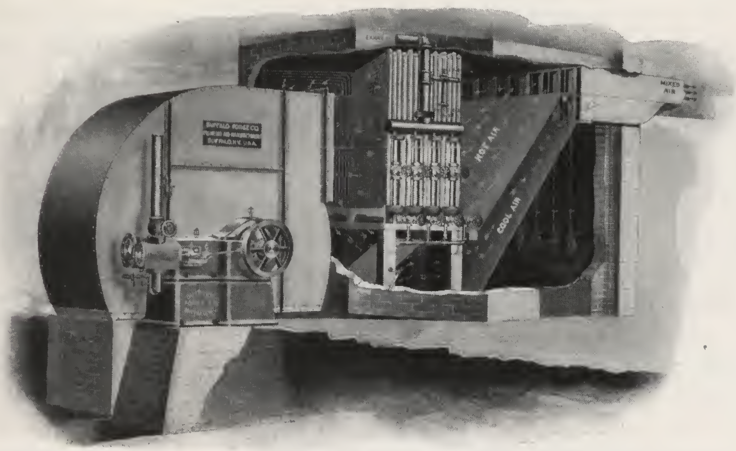
5th. The Buffalo Air Economizer, pages 50 to 52, which is an important factor in utilizing the waste heat of boiler flue gases, etc. These features, while novel have been thoroughly tested and are at present obtaining most satisfactory results in actual operation.

PART ONE

HEATING AND VENTILATING PUBLIC BUILDINGS

Foreword

It may seem at the outset that any exposition concerning the necessity for proper heating and ventilation for public buildings is superfluous. Much has already been written upon the subject. Its importance is insisted upon in theory (though often not in practice) in every schoolroom. The agitation for better ventilation in schools has assumed the importance of a crusade, and often the sacrifice of health has been due to ill-advised application of incorrect principles. Let it be the object of the following pages not only to emphasize the importance of good ventilation, but to show how proper ventilation may be secured, and to describe such apparatus as may be necessary to its attainment.



THREE-QUARTER HOUSING FAN, LEFT-HAND TOP HORIZONTAL DISCHARGE, BLOWING AIR THROUGH AND UNDERNEATH HEATER INTO BRICK PLENUM CHAMBER

While it is true that natural means are sufficient under natural conditions, it is equally true that artificial means are a necessity under artificial conditions. The tepee of the Indian, with its open fire, was sufficient for his needs and if not up to modern standards of heating and ventilation from a standpoint of convenience and comfort, it was certainly superior in regard to sanitation. With

the growth of civilization there has been a development in the methods of heating, starting with the open fire and passing through the period of fireplaces and stoves to the modern complex methods of heating with hot water, steam and hot air. Similar, although much slower and more imperfect, has been the development in methods of ventilation. We have long since passed from the stage of open windows and fireplaces to a period where artificial ventilation by means of fans has become a necessity. Indeed, in densely peopled buildings, it is a problem requiring study, experience and skill to design a system which shall provide for the entrance and the exit of air to make the ventilation complete and uniform throughout the entire building, and at the same time avoid all objectionable drafts. What, then, constitutes good ventilation and how may it be attained?

Ventilation

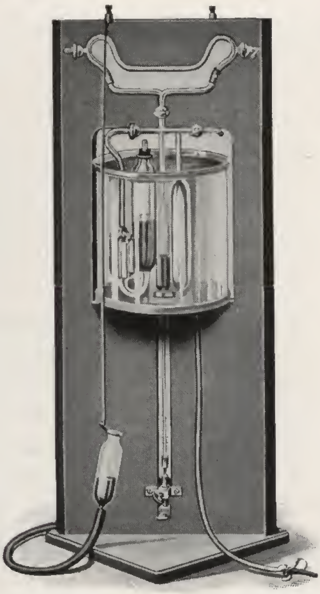
Life is sustained in the human body as in other animal organisms by a process of combustion in which the oxygen of the air is combined with the hydrogen and carbon of the assimilated food, and a new product, chiefly carbon dioxide, is formed; therefore a continuous supply of air is as essential to the sustaining of life as it is to the combustion of fuel under a boiler. Here, however, all comparison ends, for the amount of air to be supplied cannot be solved by an equation of chemical reactions, inasmuch as the "liveable" point is reached long before the chemical limit. While the percentage of carbon dioxide in the air is a trustworthy indication of the state of purity, it by no means accounts for the exceedingly harmful effects of impure air. The well known poisonous effect of respired air is believed to be due to organic matter exhaled from the lungs.

The following table gives the composition of pure air and of respired air:

	Pure Air	Respired Air
Oxygen	20.35	16.2
Nitrogen	78.10	75.4
Carbon Dioxide	0.03 to 0.04	3.4
Water Vapor	1.5 (variable)	5.

As the respired air is immediately diffused in the air of the room, it cannot be directly removed, but only diluted till it ceases to be harmful. There is therefore no definite standard of purity, and any line drawn between good and poor ventilation is arbitrary. Pure air contains three to four parts of carbon dioxide in 10,000. With an increase to eleven parts in 10,000, the air becomes noticeably oppressive, while an increase of three parts or a total of six to seven parts is scarcely noticeable.

Good ventilation is considered to exist when the air contains a total of not more than six to eight parts of carbon dioxide in 10,000. Twelve or more parts of carbon dioxide is considered exceedingly dangerous to health. It is estimated that the average adult, at rest, breathes 500 cu. in. of air per minute and exhales 17 cu. in. of carbon dioxide. From this it is possible to determine the air supply required to maintain any standard of purity as shown by the following table:



PETTERSSON'S APPARATUS FOR DETERMINING CO_2 IN AIR

CUBIC FEET OF AIR TO BE SUPPLIED PER PERSON FOR VARIOUS STANDARDS OF PURITY

Parts Carbon Dioxide in 10,000	Cu. ft. Air per Min. per Adult	Per Cent. of Respired Air
5	100	0.29
6	50	0.58
7	33.3	0.87
8	25	1.15
9	20	1.45
10	16.7	1.74
11	14.3	2.03
12	12.5	2.32

SPECIFICATION OF USUAL AIR SUPPLIES, PER PERSON

	Cubic Feet per min.
Hospitals (ordinary)	35 to 40
Hospitals (epidemic)	80
Workshops	25
Prisons	30
Theaters	20 to 30
Meeting halls	20
Schools (per child)	30
Schools (per adult)	40

Immediately associated with the problem of ventilation, is that of satisfactory heating. While the former applies chiefly to public buildings and densely peopled workshops, the latter is (outside of the tropics), a universal requirement equally necessary in buildings for all purposes and of all types. This requires the discussion of heating requirements and systems under two separate classifications, the first involving buildings requiring heating with ventilation, and the second, buildings requiring heating without ventilation.

Room Temperatures

The physical principles involved in heating buildings are much more complex than usually supposed, and exhibit an admirable nicety in the balancing of forces. We have first of all to consider

the heat generated by the human body, and the method of its disposal, which are important conditions determining the most desirable room temperatures, and in densely peopled buildings, largely determining the result of vital processes dependent in part upon the activity of the individual.

The following shows the amount of bodily heat diffused:

Child six years old	240	B.T.U.	per	hour
Adult at rest	380	"	"	"
Adult at work	500	"	"	"
Man 30 years old in an atmosphere with a temperature 68° F.	440	"	"	"
The same, in atmosphere of 31° F.	600	"	"	"
Woman 32 years old	480	"	"	"
Adult in old age	360	"	"	"

The amount of heat in B. T. U. usually assumed as given off per person per hour in an atmosphere of 70° F. is 400 for adults and 200 for children. These are the figures generally used when the heating effect of the occupants of assembly halls or auditoriums is taken into account.

Since the temperature of the adult in health is constant at about 98° F. the heat must be disposed of as fast as generated. This is accomplished in three ways: first, by radiation or direct transmission to the surrounding air, second, by the absorption of heat in the evaporation of perspiration, and third, by the evaporation of moisture from the lungs. Radiation depends chiefly upon the difference in temperature between the body and the surrounding air, but is also affected by the amount of clothing and the humidity of the air. It is evident that the temperature of the room should not be so low that the body will radiate more heat than produced under normal conditions.

In fact, from a hygienic standpoint it should be somewhat above the point of equilibrium, allowing part of the heat to be absorbed by the perspiration. The following are the usual room temperatures required.

Public buildings	68° to 72°
Machine shops	60° to 65°
Foundries and boiler shops, etc.	50° to 60°

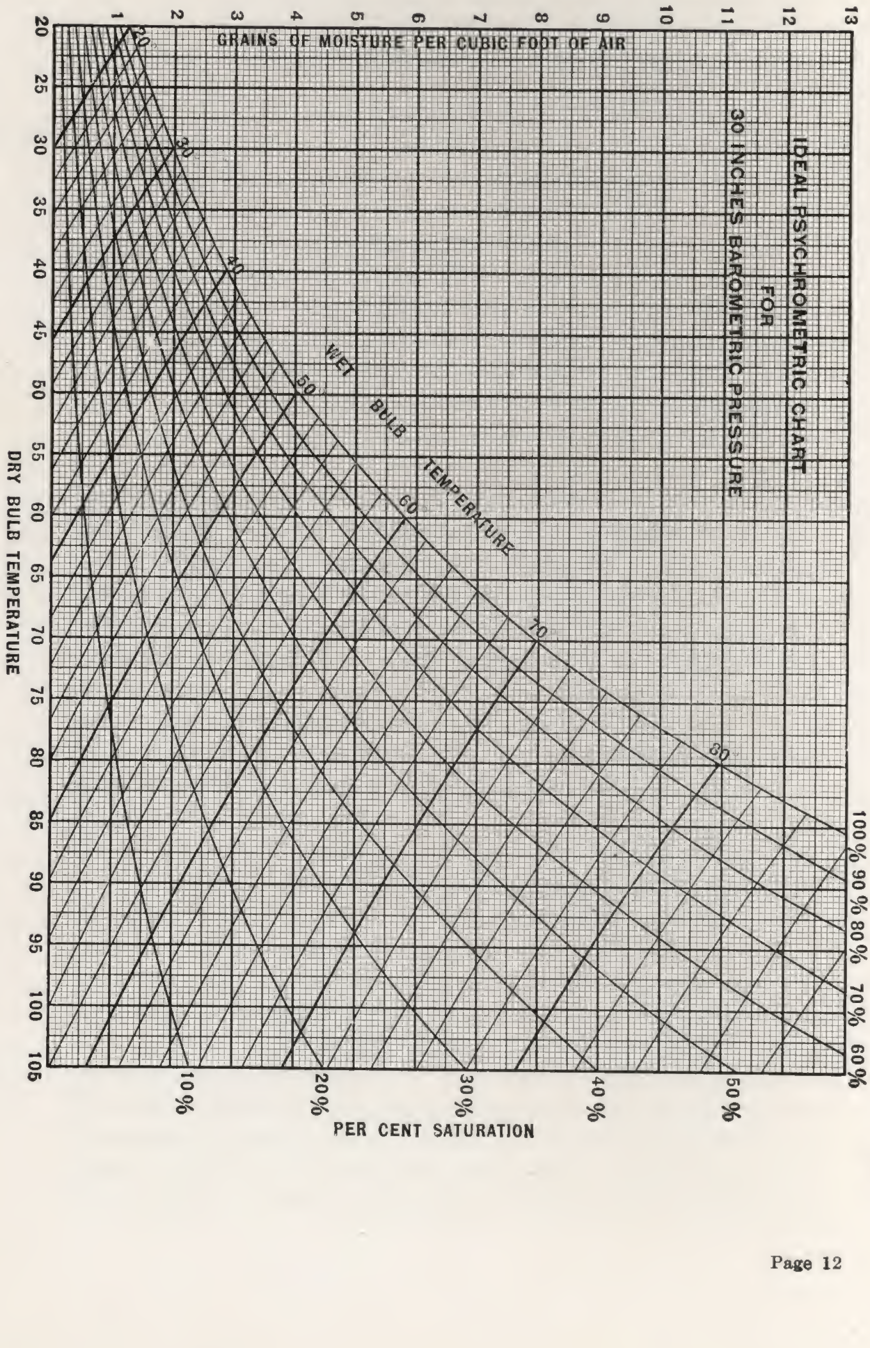
Heat Losses To maintain this temperature, artificial heat must be supplied in sufficient quantities to compensate for the heat transmitted through the walls of the building and to heat the outdoor air supplied for ventilation to the rooms. The subject of heat transmission in buildings has been so thoroughly investigated by Peclet and

others, that the laws governing it and the factors for transmission to various building materials and thicknesses of walls are now quite accurately known. It is generally assumed that the loss by transmission is proportional to the difference of temperature. The table on page 116 generally shows the accepted factors for heat transmission.

Humidity

In schools and other public buildings, the humidity of the air is of more consequence than is usually supposed. The amount of moisture which air can hold at saturation per unit of volume increases very rapidly with the temperature as shown by the psychrometric chart on the following page. At 70° it will hold 8 grains of moisture per cu. ft. while at 32° it can hold but 2 grains per cu. ft. and at zero only 0.5 grain. Air normally has a humidity varying from 50 to 70% of saturation, while if much above or below these limits it becomes uncomfortable if not actually injurious to health. Hence air at 70° should contain from 4 to 5.5 grains per cu. ft. while at 0° it contains only about 0.3 grain and at 32° 1.25 grains, so that in the usual systems of heating, while at 32° outside, the humidity of the air when heated to 70° would be only 15.5% or less than the humidity of the dryest climate known. It is this extreme dryness of the air in a heated room which produces many of the discomforts commonly noticed, but not fully explained, such as extreme thirst, a parched feeling in the nose and throat, lassitude and headache. The effect of this extreme dryness is doubtless very harmful to the mucous membrane in nose, throat and the lungs and may be considered a contributing source of many throat and pulmonary diseases.

From the hygienic standpoint it is evident that the means for regulating the humidity should be considered side by side with proper ventilation in every school or other public building. The means provided for supplying and controlling the humidity in such buildings is fully explained on page 20. The effect of humidity is also quite marked in other ways and deserves consideration from an economical standpoint. If a thermometer bulb be covered with a damp cloth as in the hygrophant, a drop of several degrees in temperature will ordinarily result. This wet bulb temperature may be termed the "sensible" temperature as it is the temperature which the body feels. This sensible temperature is dependent upon the temperature of the air and upon its humidity. The less the humidity, the greater is the difference between the actual and the sensible temperatures, while at saturation they are the same. The psychro-


IDEAL PSYCHROMETRIC CHART
FOR
30 INCHES BAROMETRIC PRESSURE
GRAINS OF MOISTURE PER CUBIC FOOT OF AIR
WET BULB TEMPERATURE
DRY BULB TEMPERATURE
PER CENT SATURATION
10%
20%
30%
40%
50%
60%
70%
80%
90%
100%

metric chart on page 12 shows the lines of equal sensible or wet bulb temperatures corresponding to different actual temperatures and humidities. It should be noted that it requires a much higher temperature in dry air than in moist air to maintain a given sensible temperature, for instance 65% humidity gives a sensible temperature of 58° or the same sensible temperature as produced by an actual temperature of 75° with 30% humidity, corresponding with the indications of the thermometer, while rooms heated to an apparently high temperature are often uncomfortably cool. While considerable heat is unavoidably absorbed in the introduction of additional moisture in the air, it may be shown that this is entirely offset, so far as ventilation is concerned, by the economy in the lower room temperature required. As a lower room temperature means a decreased radiation loss, there will be also a distinct gain in this respect.

Methods of Heating and Ventilating

One of the earlier, and in some respects one of the best methods of heating and ventilating is the old fashioned open fireplace. The draft produced by the large chimney and open fire gives ample ventilation, but the distribution of heat is necessarily poor and as a heating system is most uneconomical. The next step in the evolution of the art was the stove, which improved the economy but sacrificed ventilation. However, its modification, the hot air furnace, gives a certain measure of ventilation which nevertheless is too limited and unreliable to make its use permissible in large or densely peopled buildings. In addition to this it is often found harmful to health, owing to the unavoidable leakage of products of combustion. Owing to the low furnace temperatures permissible its economy is much below that of steam and hot water systems. The next advance is marked by the introduction of direct radiation using steam or hot water furnaces, which affords some economy of operation, because it provides for no ventilation other than by windows. Owing to its cheapness, it has been extensively introduced and as a result close, stuffy, steam heated schoolrooms, offices and public buildings have doubtless increased materially the world's death rate.

The use of indirect radiation permits a certain amount of ventilation and elaborate systems have been devised on this basis. Aspirating shafts for removing foul air in connection with indirect systems have given positive results. In the latter systems, radiators are placed in the ventilating flues to produce a draft by increasing

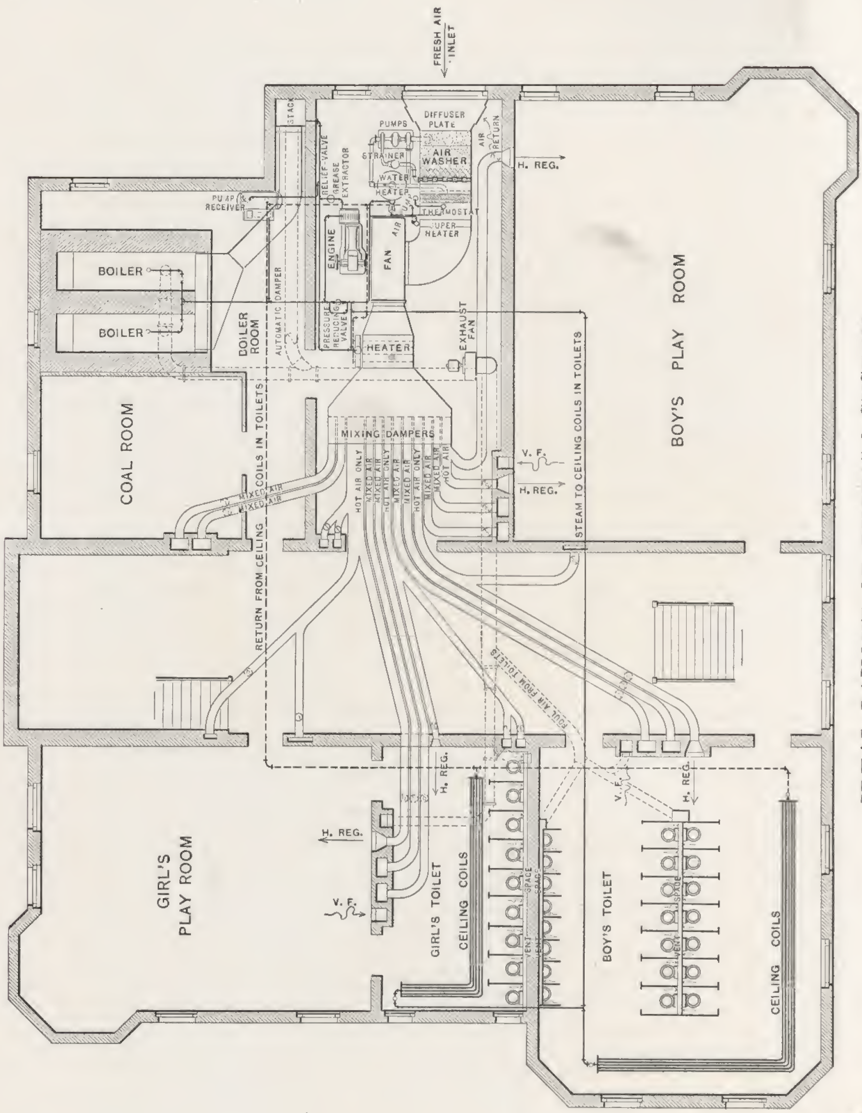
NOTE—The chart on page 12 gives wet bulb depressions corresponding to U. S. Government tables. These vary 2% from the true adiabatic curves.

the temperature of the foul air The cost of ventilation by this method is enormously expensive and the use of aspirating flues as substitutes for fans is indefensible.

The Buffalo Fan System

It is, today, a generally admitted fact that the fan system has solved the problem of the successful heating and ventilation of public buildings. In recognition of this, the legislatures of several states have passed statutory laws requiring the use of the fan system of ventilation in school buildings. Among the states which have enacted such laws are: New York, Massachusetts and Pennsylvania. The question is not, "Shall the fan system be used?" but "How may it best be applied?" For the past 30 years the Buffalo Forge Company has been engaged in designing heating and ventilating systems and in the construction of such equipment. This company has its systems in successful operation in thousands of buildings through this country and in Europe; in fact, in all parts of the civilized world. Recent improvements have been added which have brought the art of heating to a degree of perfection not previously known. One of the most important of these is the Carrier Air Washer and Humidifier, which removes all impurities from the air and imparts to it the proper humidity. There are various modifications of the Buffalo Fan System as applied to public buildings, but the most complete and most highly perfected system is known as the Buffalo Single Duct Plenum System with Thermostatic Control. The apparatus is shown on page 7. In general, the equipment required includes a boiler for the generation of steam, a centrifugal fan driven by an engine or motor for the propulsion of air, an arrangement for purifying and humidifying, a steam radiator for heating, a system of ducts and registers for the distribution of heated air, a system of vent ducts and registers for the removal of foul and cold air, and thermostatically operated mixing dampers for regulating the temperature. The boiler may be of any customary type and may be operated at any pressure between ½ and 100 lbs., however, a pressure of 20 pounds is desirable as it enables a steam engine to be employed. The fan is of the centrifugal type and is usually made an exhauster, i. e., with one inlet and is commonly driven by an engine or motor. On the score of economy the steam engine is preferable, since the utilization of exhaust steam in the heater reduces the cost of power required to operate the fan to practically nothing. Buffalo Low Pressure Engines constructed especially for this service are shown on page 38.

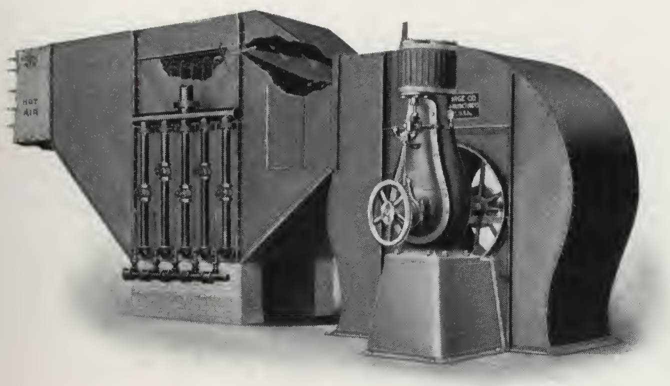
The Buffalo heater described in detail on pages 73 to 81 consists of vertical coils of 1″ full weight steel pipe screwed in cast iron manifold bases. The steam is supplied to the coils at one end of the manifold, and the condensation is removed from the other end. Separate steam and exhaust connections are provided for each of the several sections into which the heater is divided. These are supplied with valves so that as many or as few sections as desired may be used. The fresh air from outside is either forced or drawn through the stacks or coils by the fan. In public buildings the heater is usually separated into two groups, one part known as the tempering coils and containing from 6 to 10 rows of pipe which serve to heat the air to a temperature of about 60° to 70° before entering the fan, the other part, known as the heater proper and containing from 12 to 20 rows of pipe is placed at the fan outlet. Between the tempering coils and the fan is placed the Carrier Air Washer and Humidifier. The air after being tempered, cleaned and properly moistened is discharged under a slight pressure (about ⅓ oz. per sq. in.) into two chambers known as the hot and tempered air plenums, respectively. In the hot air plenum chamber are placed the heater coils, while the supply to the tempered air is carried by a by-pass either underneath the heater as shown on page 7, or over the heater as shown in the apparatus on page 17. Leading to each room of the building is a single duct which connects with both the hot air and tempered air chambers. The air supplied to each of these ducts is controlled by a double mixing damper illustrated on page 18. This damper is governed automatically by a thermostat placed in the room, which serves to regulate the proportion of hot and tempered air to suit the requirements of the room without affecting the total volume of ventilation. The horizontal distributing ducts in the basement are of galvanized iron. These connect with vertical ducts built into the walls which convey the air to the rooms on the different floors. The arrangement of hot air and vent registers is well shown on page 18. They are both preferably placed on one of the inner walls of the room. The heated air enters the room at about eight feet above the floor which is sufficient height to prevent drafts, with proper air velocities through the registers. The cold air is removed by vent registers placed at the floor line usually on the same side of the room as the hot air flue. The warm air enters above at a higher temperature than that of the room, and remains in the upper stratum where a complete and practically uniform diffusion to all parts of the room occurs by reason of its lighter



IDEAL LAYOUT OF BUFFALO SYSTEM

weight. The cooling effect of the outer walls and windows produces a downward circulation at these points with a consequent flow from the hot air registers toward the outer wall in the upper stratum, and a flow from the outer walls to the vent registers in the lower or breathing stratum. The flow occurs over such large areas that the velocity is imperceptible.

As the air is positively supplied to the room, it must be positively forced out through the only channels available, the vent registers, flues and cracks about the windows and doors in the outer walls. Exit by the latter means is necessarily restricted in a properly constructed building, but it serves the very useful purpose of preventing the undesirable infiltration of cold air which would otherwise occur.

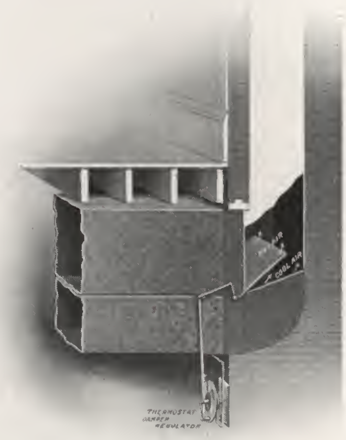


RIGHT-HAND TOP HORIZONTAL DISCHARGE FAN, BLOWING AIR THROUGH HEATER, CONNECTED TO VERTICAL ENGINE

The above forced method of ventilation is commonly designated as the plenum system. However, in well designed plants it is not dependent upon the forcing action alone to provide ventilation. The vent flues, being vertical and containing air of a higher temperature than that outside, act as chimneys, and produce ample draft for the purposes of the plenum system when properly designed.

In the exhaust or vacuum system of ventilation, the air is positively removed by means of exhaust fans connected with the vent flues. Little can be said in defense of the vacuum system when

employed alone, but used in connection with the plenum system it often proves a very desirable adjunct. It enables the use of smaller vent flues and renders the draft positive and equal at all times, independent of external temperatures. However, there is a danger in exhausting the air too rapidly, as then a vacuum is produced in the room which causes an inward leakage of cold air. This disturbs the uniformity of the room temperature and increases the cost of heating.



BUFFALO AIR MIXING DAMPER
CONTROLLED BY THERMOSTAT

Other Systems

There are several other arrangements of the Buffalo system, which, while not so perfect as the above, are sometimes better adapted to certain conditions. The Buffalo Double Duct System with hand regulation as shown on this page, has the same general arrangement of apparatus as the system previously described, the only difference being that two (2) horizontal distributing ducts are used in place of one. One of these is for hot air and the other for tempered air. At the point where they enter the flue a mixing damper is placed which is operated by a chain leading to the room. Raising or lowering the damper increases or decreases the proportion of hot or tempered air. The temperature of the room is thus readily controlled at pleasure. This system is well adapted to schools and offices when the additional expense of thermostatic

HAND OPERATED MIXING DAMPER
FOR HOT AND TEMPERED AIR

control is not desirable. Thermostatic control can subsequently be readily installed if required.

It is sometimes preferred to use direct radiation entirely for heating and depend upon the fan system for ventilation only. In this case the heater coils are omitted, and the air supply is maintained at the room temperature either by hand or by automatic regulation. An objection to this system is that the first cost of installation, and also the operating cost, is considerably higher than the usual fan type system. Another serious objection is that the operating engineer will see that the building is properly heated, but may not bother himself to get proper operating conditions on the fan, simply being satisfied to see that it is turning over, and consequently ventilation will be poor.

Advantages of The Buffalo Fan System

Nothing could be more striking than the contrast between the methods and effects of the old system of direct radiation with windows for ventilation, on one hand, and the Buffalo Fan System of Heating and Ventilation on the other. Less marked, yet considerable is the contrast between indirect gravity systems and the fan system. With direct radiation all air for ventilation must be admitted at the windows through the lower sash. This is at once most unsanitary and uneconomical. Unsanitary; first, because in crowded rooms it is impossible to admit sufficient air in this way; second, the ventilation is not uniform; and third, unbearably cold drafts are produced and an undesirable layer of cold air settles along the floor, thus doing far more harm than an entire lack of ventilation. It is not economical because the coldest air remains along the floor, and the heated air immediately rising and flowing out of the window openings, the heat produced is not used. It is also wasteful because the heat is not equally distributed, the better ventilated parts of the room are too cold, the poorly ventilated parts are too hot; the room temperature is not easily regulated and loss by overheating is commonly great.

The Buffalo Fan System, on the other hand, is sanitary and economical. Sanitary—because it removes positively all foul and disease laden air, which authorities maintain is the cause of 40 per cent. of all mortality—because it maintains a uniform temperature, prevents all drafts and secures a warm floor. Economical—because it retains all the warm air and removes the cold air at the floor line, because with it the temperature is readily and absolutely controlled

either automatically or by hand, and overheating is prevented. This latter advantage is much greater than generally supposed.

Objection to the use of the fan system exists in some places on account of dust drawn in with the air and blown into the room. This has been entirely overcome by the air washing apparatus, as all traces of dust, soot, and smoke are now removed and the foulest germ laden air of the city can be made as clean and pure as that of the country. The advantage of this process wherever cleanliness and sanitation are desired is easily appreciated and renders the system particularly valuable in libraries, hospitals, hotels, schools and other public buildings. The advantage of affording a proper humidity is quite as important. Under the subject of humidity on pages 11 to 13 the insufficiency of moisture in the air of heated buildings is shown and the consequent injurious effects explained.

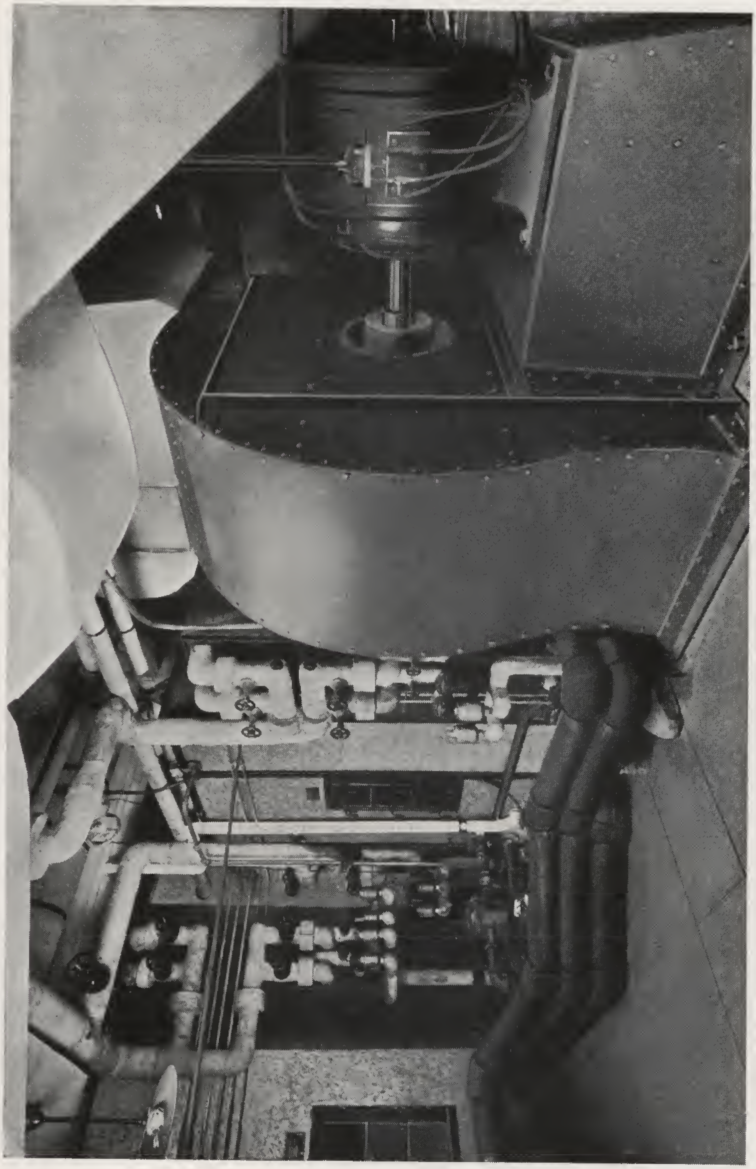
Humidity Control

The Carrier Humidifying system overcomes this objection entirely and places the humidity of the air under accurate and automatic control. It will be noted that by control of the temperature of the air entering the spray chamber through the agency of the tempering coils and by-passes, and particularly through the control of the temperature of the spray water, the humidity of the air entering the ventilation system is regulated to the finest nicety. The temperature of the spray water is raised by the introduction of steam through a device similar to an injector. It may be stated that it has been found from test that the temperature of the water is a greater factor in the amount of moisture which the air will absorb than is the temperature of the air itself, and that the relative humidity produced by the water sprays is never higher than is desirable unless the spray water is heated. This is true regardless of the outdoor temperature and humidity.

If in public buildings or industrial applications it is desired to cool the air handled and condense from it the larger portion of the moisture which it already contains, this may be accomplished by lowering the temperature of the spray water by placing brine coils from a refrigerating machine in the settling tank. In some instances the refrigerating machine is unnecessary, such as in dry kiln work where it is desired to return the air to the kiln with a fair percentage of moisture, to guard against too rapid surface-drying.

Application to Schools

Modern school buildings offer most exacting requirements in heating and ventilation. On account of the large number of pupils seated in one room, a very rapid air change is required, and this



PLANOIDAL STEEL PLATE FAN INSTALLED IN HOTEL McALPIN, NEW YORK

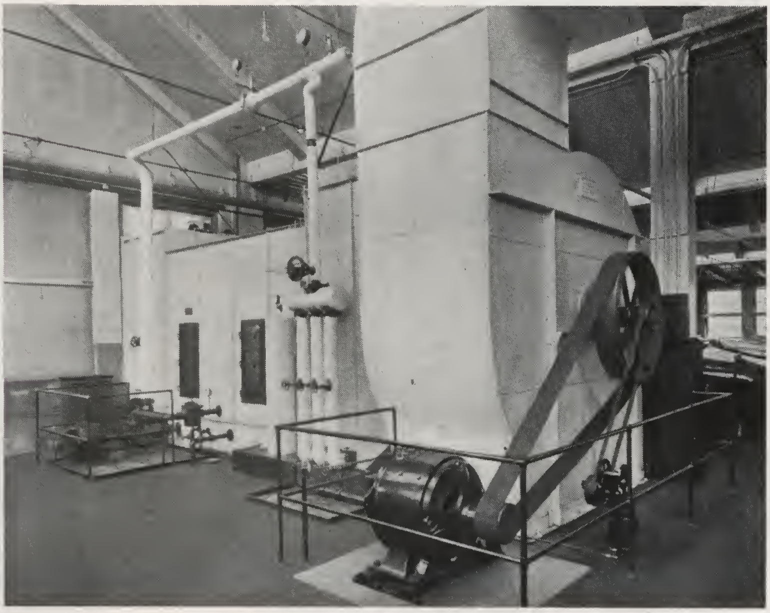
must be accomplished without drafts. The temperature must be uniform everywhere, and ventilation must be adequate. Owing to the peculiar adaptability of the Buffalo Fan System, the former is easily accomplished, the latter, however, is more difficult of attainment. Even elaborate systems cannot secure entirely perfect distribution of air, and the only practical and successful method of insuring ample ventilation in all parts of the room is to supply air considerably in excess of the theoretical requirements. The necessity of this added capacity, or factor of safety as it may be termed, is often overlooked in writing specifications for school buildings. Thirty cubic feet of air per pupil which is usually specified will allow from six to seven parts CO_2 in 10,000. Individually this is ample, but collectively insufficient, since to insure that this per cent. of CO_2 is nowhere exceeded, it would probably be necessary to supply an average of nearly 40 cubic feet per pupil.

Theaters and Churches

Audience halls, such as theaters, churches and lecture rooms, though in use but for a short time, are notorious for their poor ventilation. Much, however, is being done through the introduction of the fan system to relieve their disagreeable and unsanitary conditions. The problems of air distribution and avoidance of drafts are greatly increased owing to the usually large dimensions of such buildings, and to the density to which they are peopled. Two plans of ventilation are in vogue for audience halls. These are usually distinguished as the upward and the downward systems of ventilation. In the upward method the air is admitted through perforations in the floor underneath the seats and is allowed to escape through ventilators in the roof. In the downward system the air is admitted through registers in the walls at a height of several feet above the floor, and removed through vent registers in the walls at the floor line in the same manner as in school buildings. In large auditoriums the upward method is doubtless preferable when the architectural design makes it permissible. A perfect distribution of the air can be secured, and the air flow is upward in accord with natural air currents induced by the heat of the body and the breath. The products of respiration, and eliminations from the body are immediately carried away, and the incoming air is uncontaminated. This method of ventilation is exceedingly efficient, as a high standard of purity can be maintained at the breathing line with a comparatively small air supply.

Hospitals

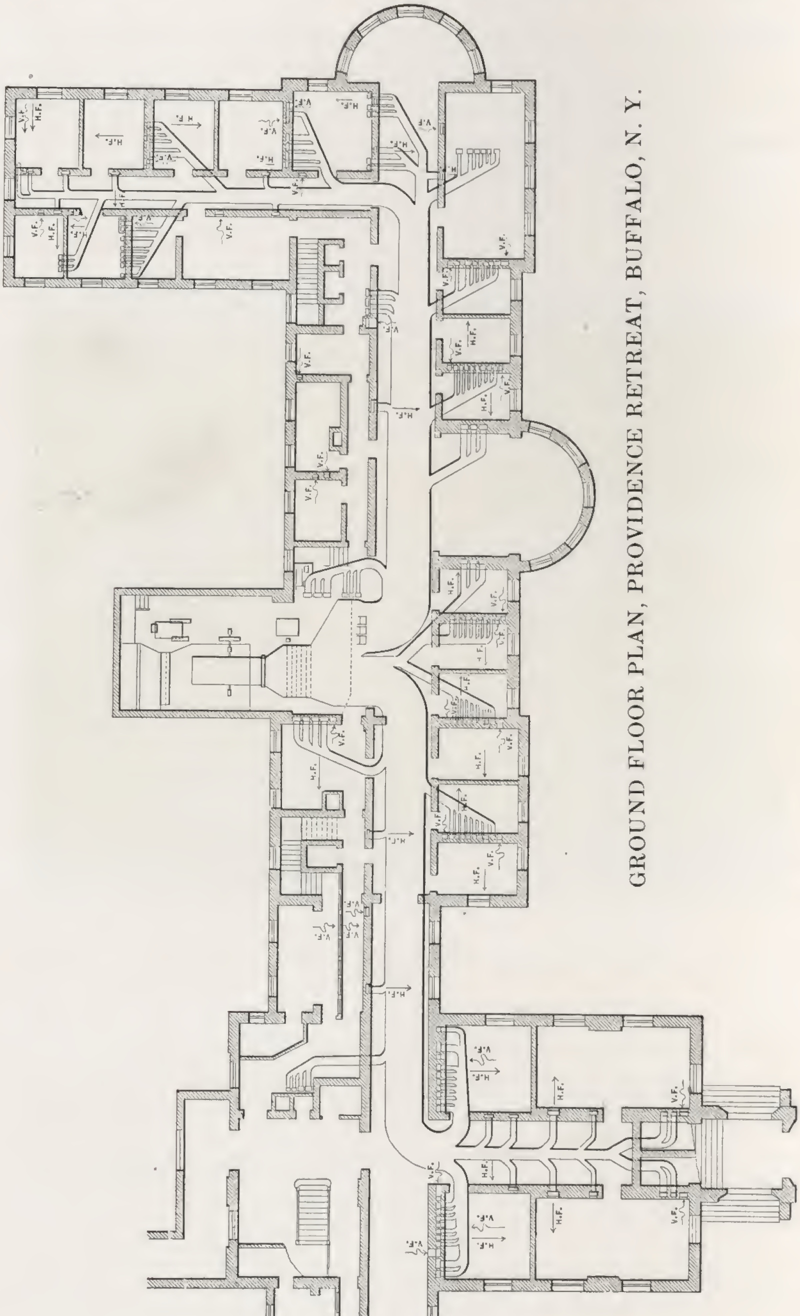
The necessity of ample ventilation in hospitals is now so well recognized that the fan system is being universally installed. The Carrier Air Washer and Cooler is a most valuable adjunct where cleanliness and heathful conditions are of paramount importance as in hospitals. In the treatment of certain diseases, the maintaining of proper humidity is especially desirable. In diseases of the



PLANOIDAL FAN AND HEATER INSTALLED WITH
CARRIER HUMIDIFIER

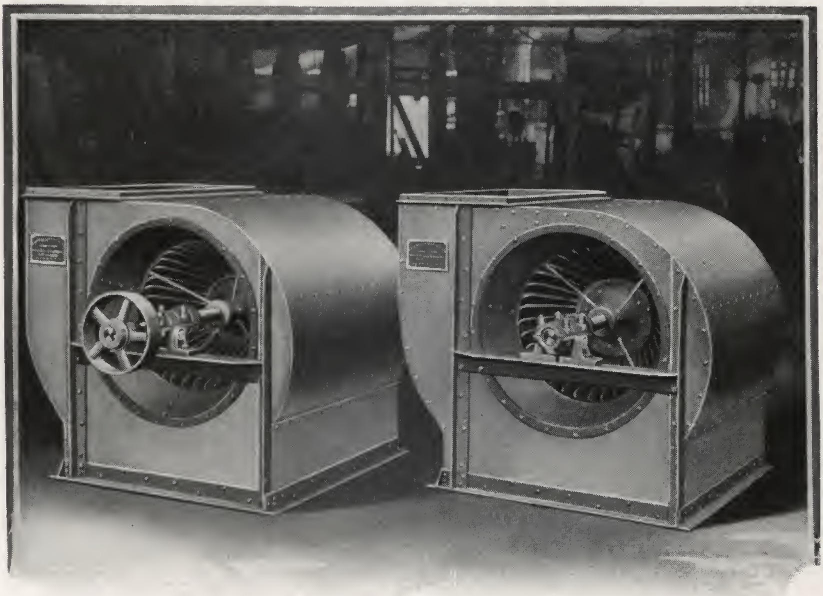
heart or of the respiratory organs, in fevers, and especially in all nervous disorders, patients are extremely sensitive to humidity conditions and adversely affected by the extreme dryness ordinarily existing in heated buildings. Used for cooling the hospital rooms in the heat of summer, the above system proves most efficacious and convenient.

The auditorium of the Cornell Medical College at Ithaca, N. Y., affords an example of upward ventilation. The floor is terraced



GROUND FLOOR PLAN, PROVIDENCE RETREAT, BUFFALO, N. Y.

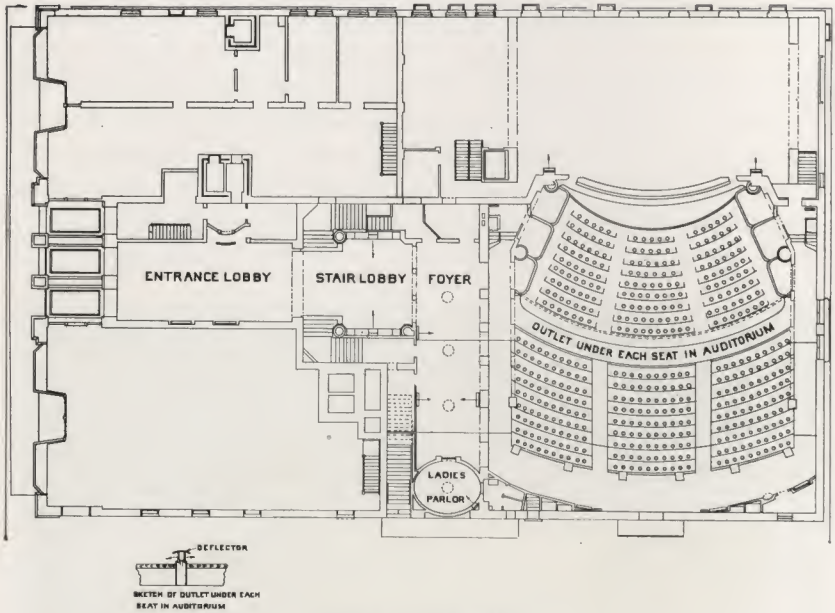
so that under each tier of seats there is a place for a series of registers. The whole space underneath the floor is made a plenum chamber and the air is discharged through registers underneath the seats at a low velocity. Another noteworthy feature of this installation is the arrangement of the air distributing system. A plenum chamber is formed by two brick partitions running the entire length of the basement. In this chamber the fan wheel is placed without housing



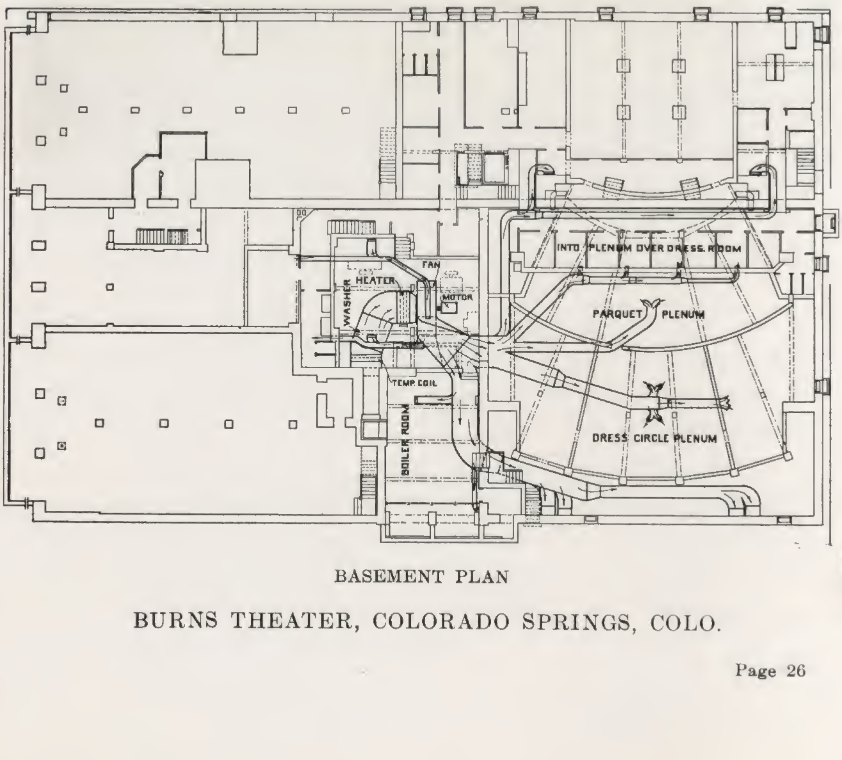
TWO DOUBLE WIDTH NIAGARA CONOIDAL FANS, FOR HEATING AND VENTILATING THE DEPEW SCHOOLS, DEPEW, N.Y.

and discharges freely in all directions. This plenum chamber connects directly with the vertical air ducts supplying the various rooms of the building.

Upward ventilation to be successful requires a very careful arrangement of the supply openings on account of the greater liability of drafts. The velocities are necessarily low, and the registers are so small that a very large number is needed to convey the necessary air.



FIRST FLOOR PLAN

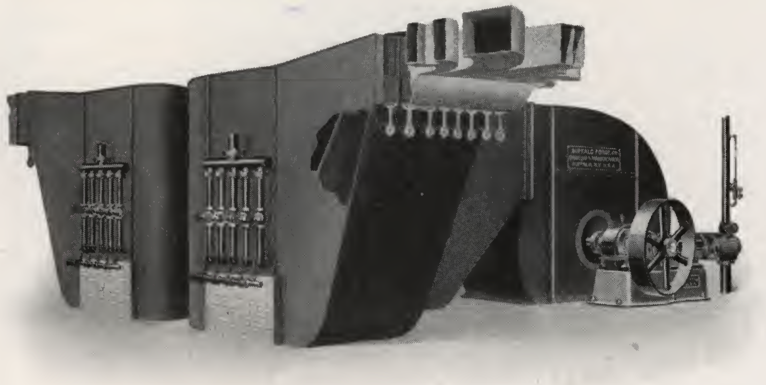


BASEMENT PLAN

BURNS THEATER, COLORADO SPRINGS, COLO.

The plenum chamber for supply is sometimes out of the question, and on this account the downward system, which is in almost universal use in schools, is extended to churches, theaters and halls with high ceilings. With a proper arrangement of fresh air and vent registers, and ample air supply excellent results are obtained. To insure such results exhaust systems are frequently relied upon, the vent registers being connected with suction fans which maintain a steady draft.

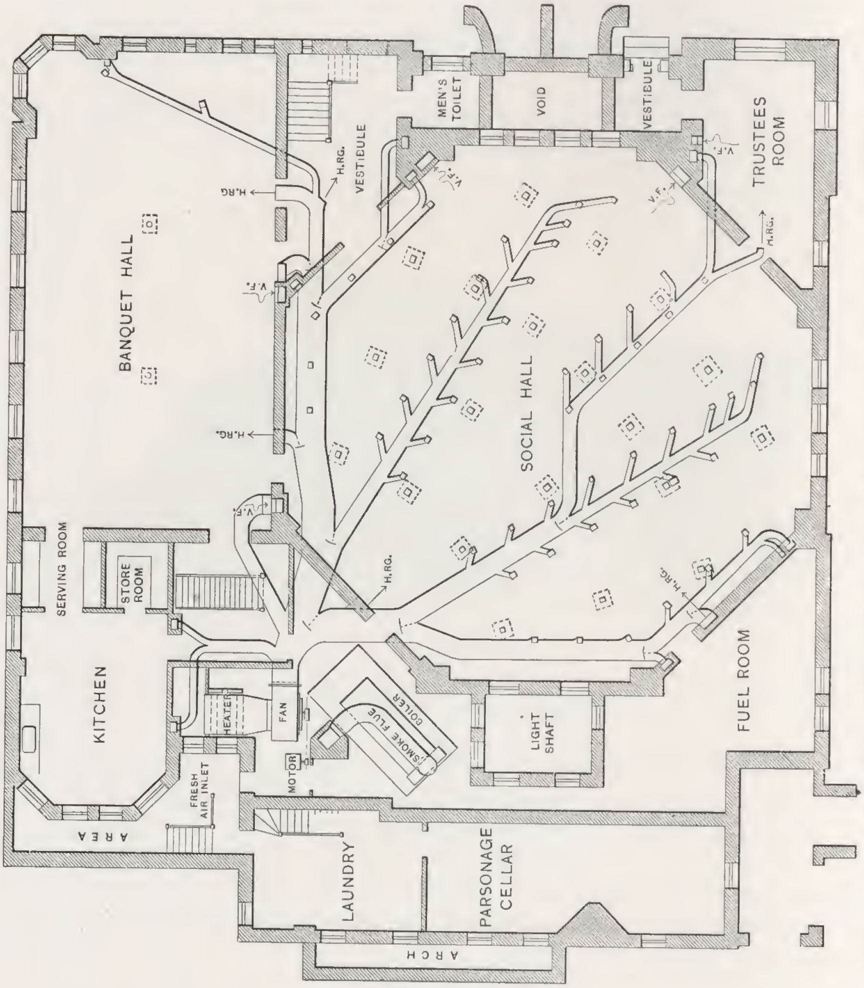
The design of theaters and churches often prevents the location of vent flues except in outside walls, the cooling effect of which seriously impairs the efficiency of the natural draft. It is always advisable to connect flues so located with suction fans.



RIGHT-HAND TOP HORIZONTAL DISCHARGE FAN, BLOWING AIR THROUGH AND UNDERNEATH MANIFOLD HEATERS

In theaters which are in use during the summer, the air washer provides the means of securing freedom from distressing heat. In order to maintain the best cooling effect, refrigerating apparatus for lowering the temperature of the water sprays is necessary, and may be economically installed and operated, but even without the use of refrigerated water the cooling effect is considerable and of decided practical value.

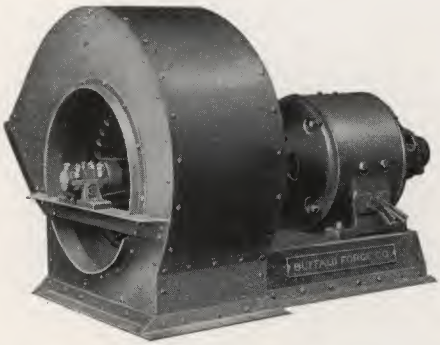
The air washer and cooling apparatus enables the temperature to be lowered about 10°, converting the theater from the most uncomfortable to the most comfortable place in warm weather, while in winter it gives a cleanliness and an increased freshness to the air supplied.



HEATING AND VENTILATING PLANT, PRESBYTERIAN CHURCH, HOMESTEAD, PA.

Libraries

The Buffalo Fan System in connection with the system of air purifying is applied to the heating and ventilation of library buildings with the most satisfactory results. Not only does it afford positive ventilation, but it frees the air from all traces of smoke and dust so objectionable in libraries; besides, the pressure of the air in the building prevents the entrance of dust from without.



NIAGARA CONOIDAL FAN DIRECT CONNECTED TO MOTOR

In one instance tests were made of the temperature of the water in circulation and of the air at various points with the results shown in the following table:

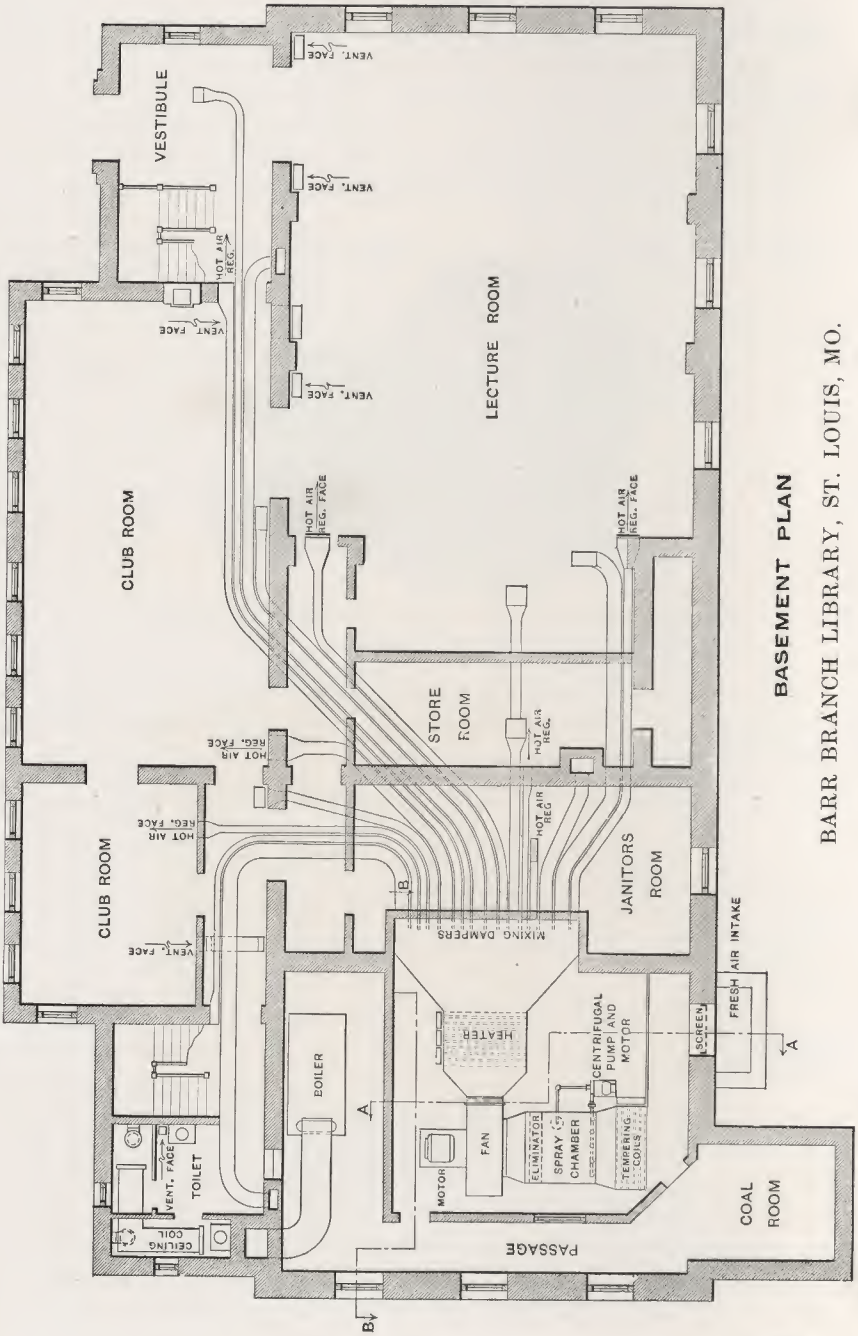
CARNEGIE BRANCH LIBRARY, ST. LOUIS, MO.

Room	Time—P. M. 2:30	2:50	3:15
Auditorium, basement	75	75	74
Stall room, basement	79	80	77½
Stall room, basement	77	77	76
South reading room, main floor	78	78	78
North reading room, main floor	78	78	78¼
Stock room	79	80	79½
Average	77.7	78	77.2

External air	86
Air entering rooms	73
Circulating water	69

It is interesting to note the effect of this apparatus in cooling the building. Although the temperature of the external air was 86° F., it entered the rooms at 73, and kept their temperature down to between 77 and 78, a cooling of about 8½°.

During the first two series of readings the windows in the three basement rooms were open. Before taking the 3:15 p. m. reading they were closed. The result shows that the temperature of these



BASEMENT PLAN

BARR BRANCH LIBRARY, ST. LOUIS, MO.

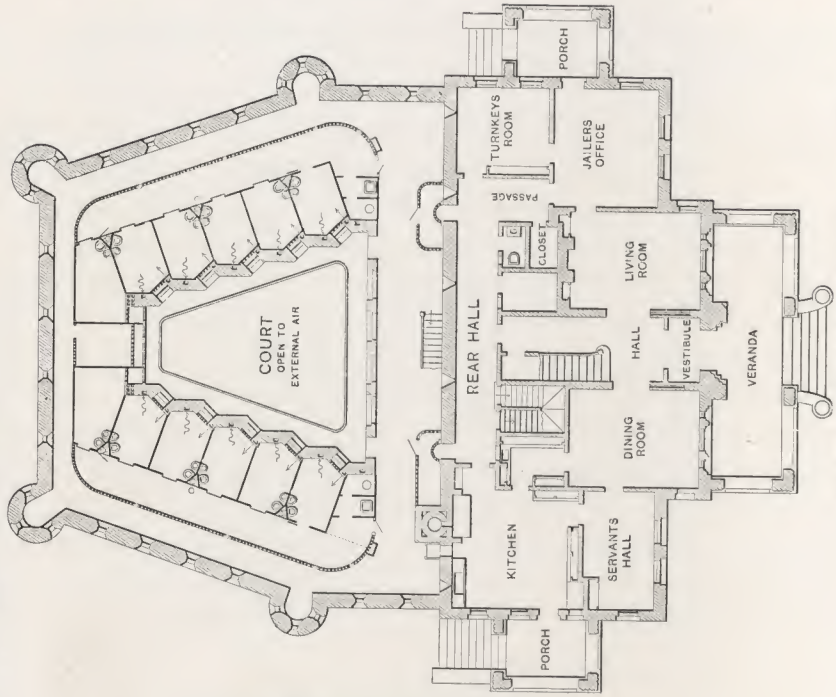
rooms was noticeably lowered by excluding the external air and supplying only the washed air from the fan.

Department Stores

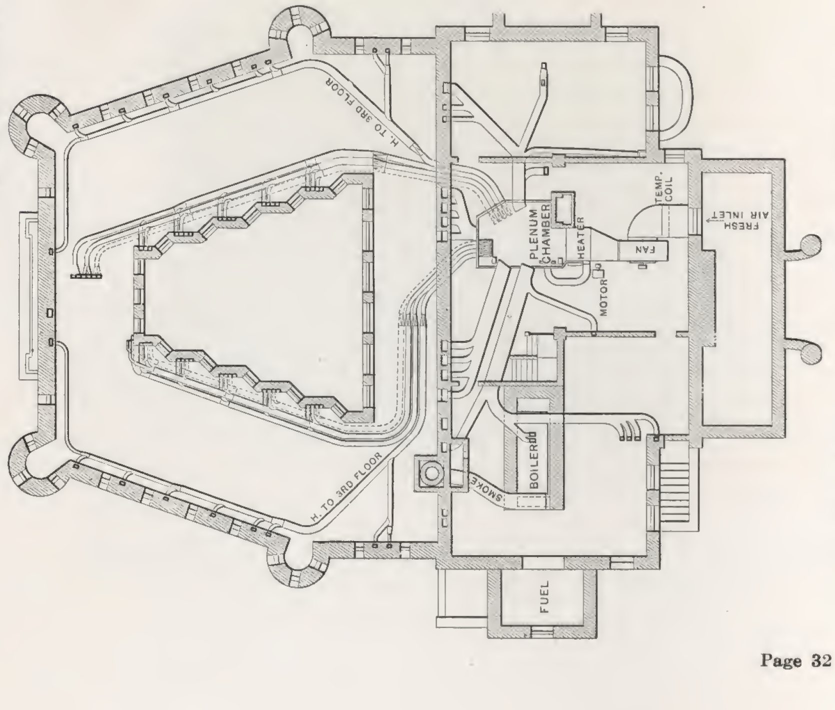
Department stores offer an especially useful field for the application of the fan system. In cold weather there exist disagreeable cold drafts along the floors. On the other hand, although on account of the crowded condition ventilation is most urgently needed, no provision is made for supplying it. The fan system fills both of these needs, first, by furnishing warmed air in large volumes without the production of drafts, and second, by creating an outward pressure which effectually prevents the entrance of cold air at the doors even when left open. The objection to the fan system previously existing on account of the dust carried into the building by the fan is entirely overcome in the Buffalo Fan System by the use of the air purifying apparatus, while at the same time the store is made very attractive in the hot days of summer by the effect which may be obtained when using this system of cooling.

SPECIAL THREE-QUARTER HOUSING STEEL PLATE PLANOIDAL TYPE "L" FAN

SUMMIT COUNTY JAIL, AKRON, OHIO



PLAN OF FIRST FLOOR

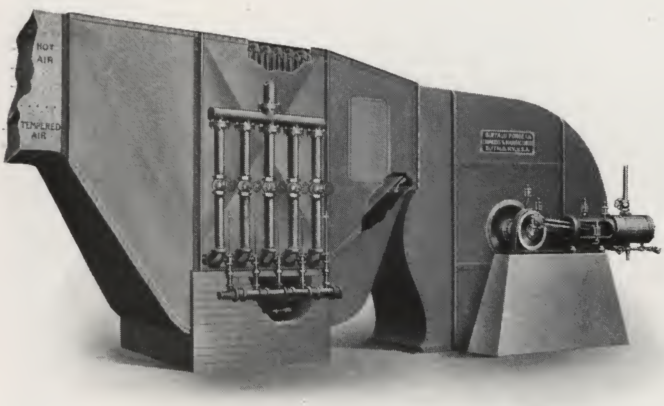


PLAN OF THE DUCTS IN BASEMENT

PART TWO

THE APPLICATION OF THE BUFFALO FAN SYSTEM TO INDUSTRIAL BUILDINGS

In public buildings, the question of heating and ventilation is decided from the view point of sanitation rather than economy. A few years ago exactly the opposite could be said of industrial buildings. It is still true, of course, that economy is a main consideration in the heating of the latter class of buildings, but sanitation is being considered more and more, which is due to two principal causes: first, that it is generally conceded that a workman's efficiency depends directly on his comfort and contentment; second, due to factory laws in this regard, which are being made more strict each year.

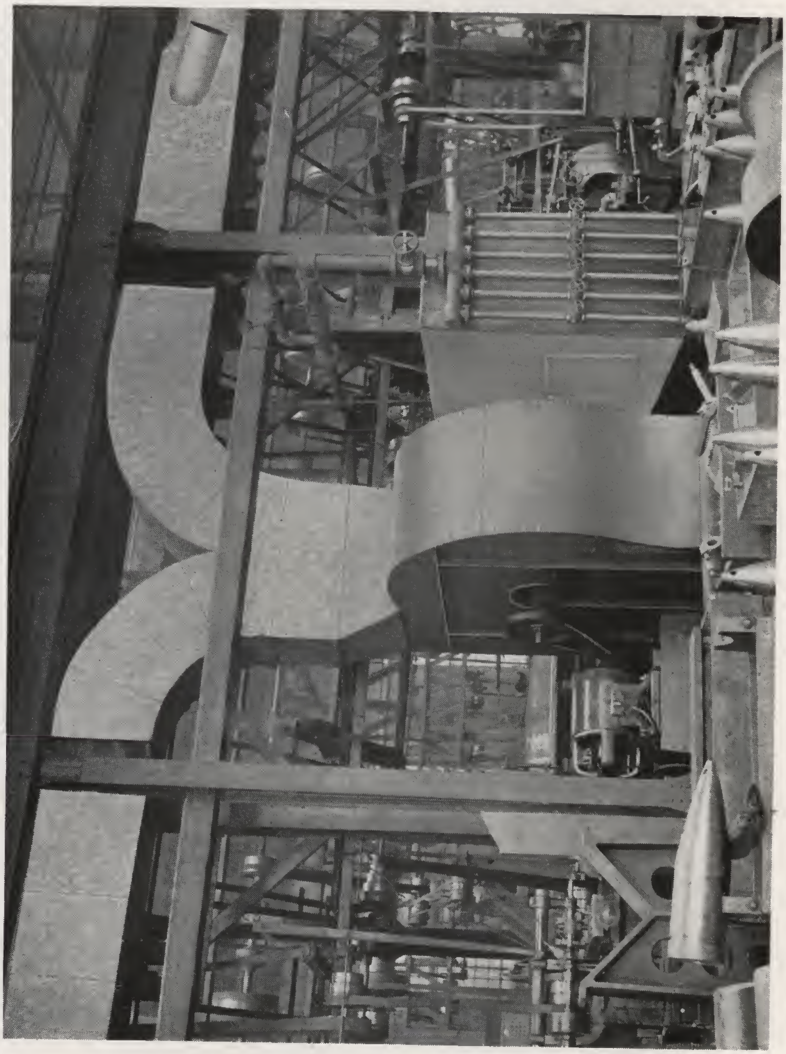


FULL HOUSING TOP HORIZONTAL DISCHARGE FAN, BLOWING AIR THROUGH AND UNDERNEATH HEATER

The Apparatus

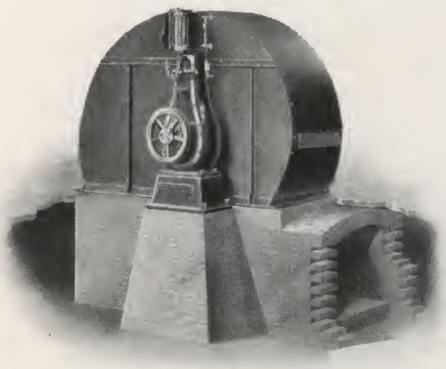
The general construction of the fan system apparatus is illustrated on pages 35 to 38, and is similar to that used in public buildings. The heating system is composed of three essential elements: —the heater, the fan, and a system of air distributing ducts.

The heater consists of rows of vertical 1-inch pipe screwed into a manifold cast iron base which is divided into separate units or sections. The coils are tightly enclosed on the top and sides by



HEATING SYSTEM IN PROJECTILE DEPARTMENT, BETHLEHEM STEEL COMPANY, BETHLEHEM, PA. ONE OF SIX UNITS INSTALLED IN NO. 2 MACHINE SHOP

a sheet steel casing, which is strengthened by angle irons around top, bottom and sides. The air is drawn or forced through between the pipes by means of a centrifugal fan which connects with the heater casing. The fan should be amply large, and be driven at sufficient speed to produce an air velocity of about 1,200 feet per minute through the clear area of the coils. This velocity is an important condition, since the effectiveness of the coils is largely dependent upon it. The increased efficiency of the heating surface from this cause is so great that only from one-third to one-fifth as much surface is required with the fan system as with direct radiation. Further, as will be shown later, the heat is so applied and distributed that it is far more thoroughly utilized than in ordinary radiation.

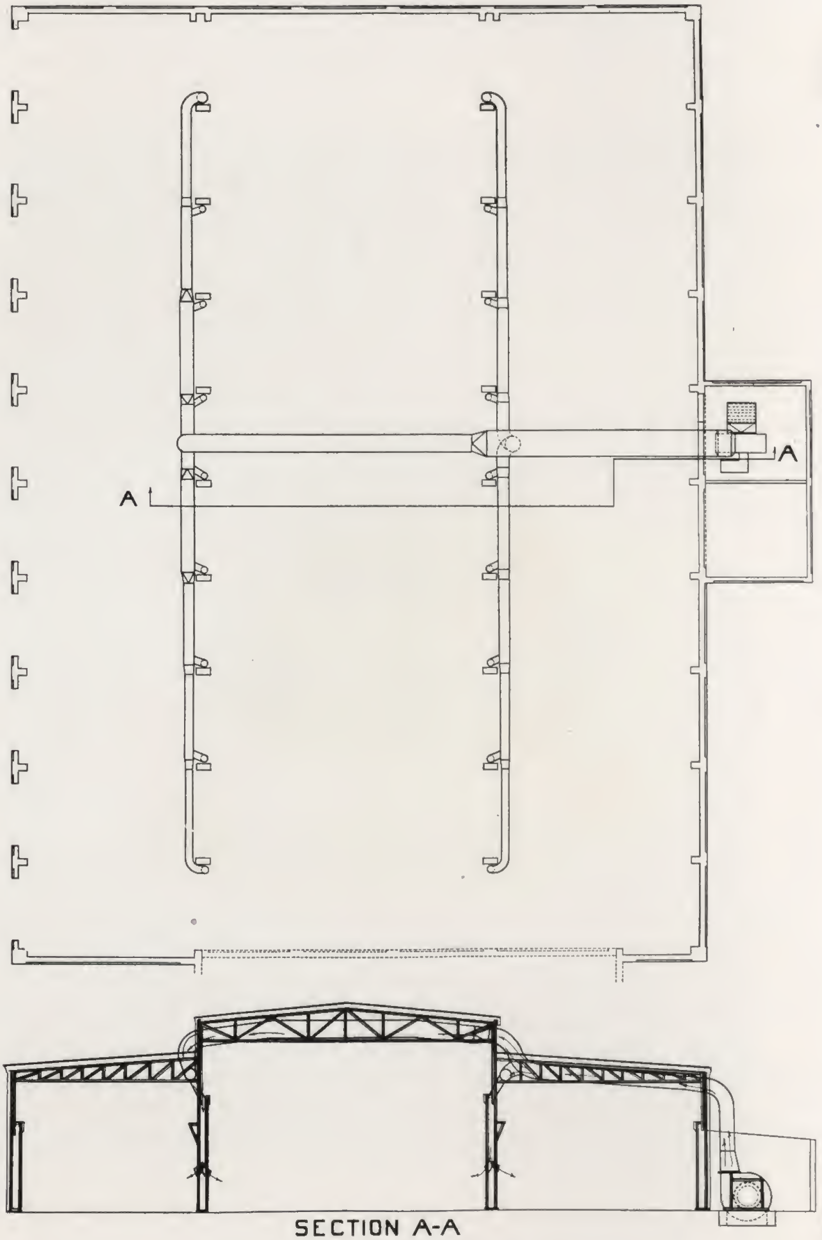


THREE-QUARTER HOUSING, BOTTOM HORIZONTAL DISCHARGE STEEL-PLATE FAN WITH VERTICAL ENGINE

Heat Losses

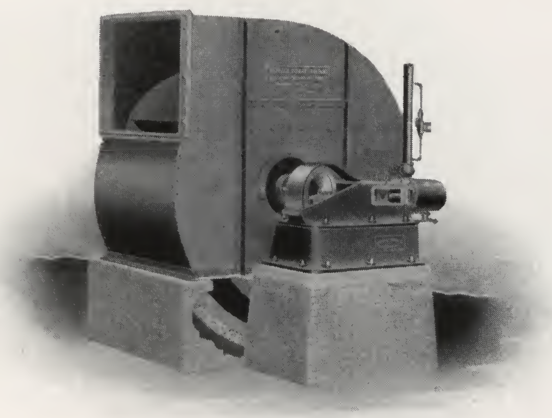
Heat losses occur in a building from two causes: first, by the direct transmission of heat through the walls and exterior surfaces of the building, and second, by the infiltration of cold air from without.

In designing a heating plant, the first of these losses may be very accurately determined by referring to tables which have been prepared, showing the amount of heat radiated under different conditions through various thicknesses of walls, windows, doors, floors, etc. The heat lost through infiltration differs so greatly in various sizes and constructions of buildings that no definite rule can be given. The allowance to be made for this is necessarily a result of experience and of careful tests of previous installations.



B. R. & P. R. R. REPAIR SHOP

This infiltration or leakage is produced by the unbalanced pressure of the column of heated air within the building, and that of the cold air without. The action is, in principle, precisely that of a chimney. The difference in pressure produced can be measured in inches of water, and increases in direct proportion to the difference in temperature between the air within the building and that without. Since the flow of air is proportional to the square root of the pressure, that amount of air entering or leaving the building through leakage will be in proportion to the square root of the difference of temperature. The effect of this leakage is as evident in theory as it is noticeable in practice. The air which escapes from the building is naturally the very hottest, and therefore, has not had its heat fully utilized; while that which enters along the floor, chills the air at the lower part of the building perceptibly, forming a cold layer of air

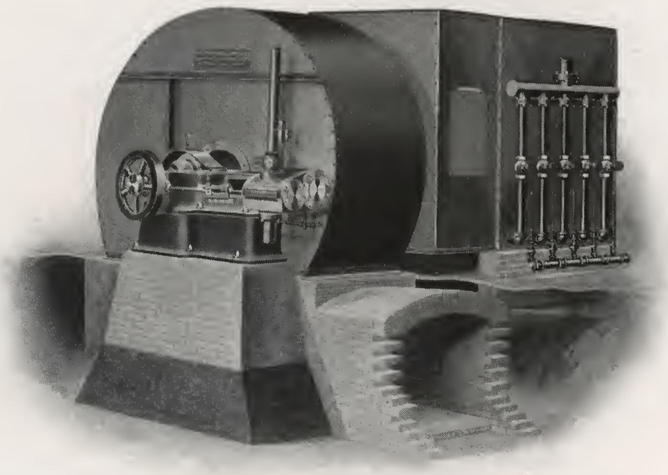


RIGHT-HAND THREE-QUARTER HOUSING, TOP HORIZONTAL DISCHARGE FAN, DIRECT-CONNECTED TO SIDE-CRANK ENGINE

which cannot be removed except by a positive circulation or diffusion with heated air such as may be secured by the fan system. In large machine shops and foundries, this layer of cold air may frequently be found to extend from 4 to 6 feet above the floor, while overhead there is a volume of overheated air which, if utilized, would heat the entire building. The most effective remedy for this evil is to maintain a slight pressure within the building by

means of a fan which takes a portion of its air from without, thereby causing a displacement and removal of cold air.

Fan System vs. Direct Radiation In either fan or direct radiation systems, difficulty is likely to be experienced from the rise of heated air which forms a stratum just beneath the roof. In machine shops and foundries, owing to their height and to the great amount of skylight surface, which is always provided in the best modern construction, the loss occasioned by this action of the heated air may be considerable, and its prevention is a serious problem. In direct radiation, where the air currents are wholly due to the difference in temperature, the attendant loss, which is relatively great, is unavoidable. Practically the only way in which this heated air can be made use of is by placing the coils next to the wall near the floor, and allowing the heated current to



LEFT-HAND BOTTOM HORIZONTAL DISCHARGE PLANOIDAL STEEL-PLATE FAN, DRAWING THROUGH HEATER

pass upward along the walls, but even this method is wasteful from the fact that it heats the walls unduly, causing a loss which may usually be estimated as great as 25% of the total heat supplied. With the fan system, however, since the method of distributing the air is entirely mechanical, it affords an opportunity for utilizing its heating effect to the very best advantage. Various methods of distribution have been devised with the Buffalo Fan System whereby

the effect of a rising current of heated air is almost entirely avoided. These systems, in general, depend upon securing diffusion of the heated air along or near the floor line.

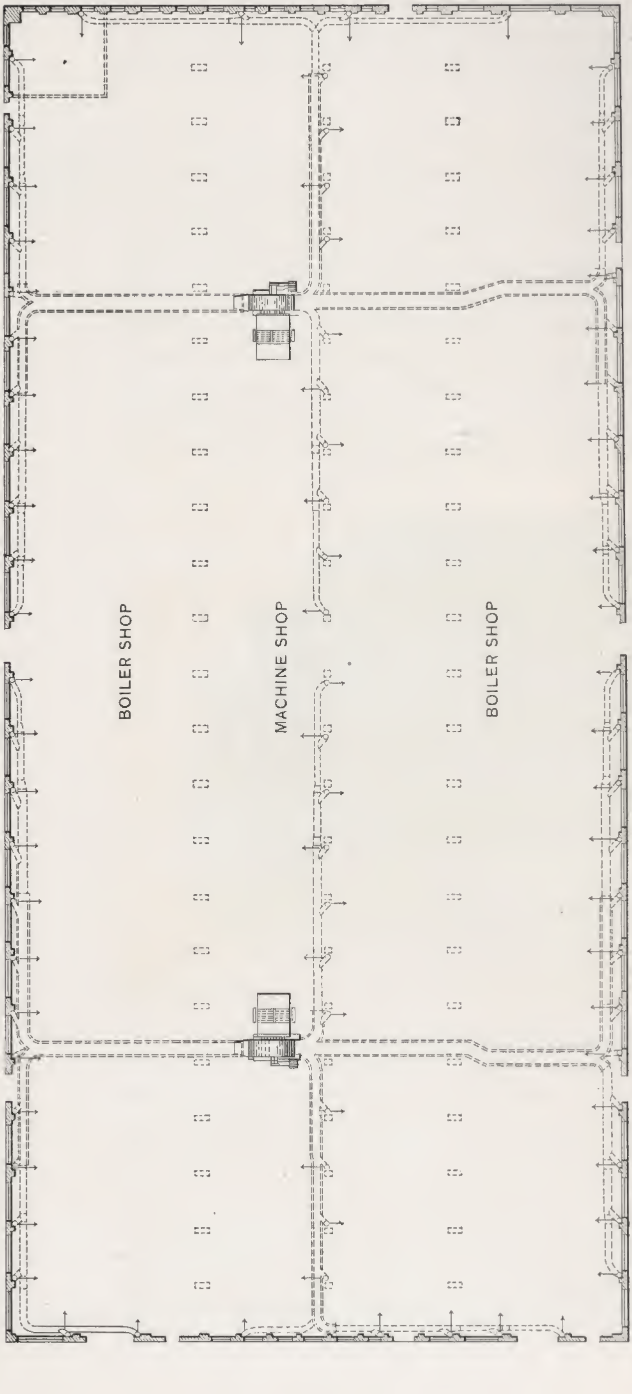
Systems of Air Supply

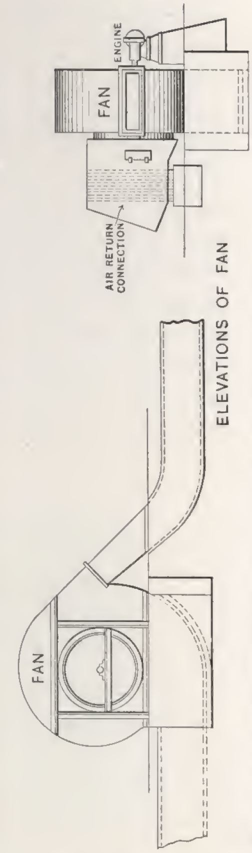
The method of distributing the air in the building is a consideration of chief importance. The usual methods of supplying heated air are: first, to take the air entirely from without and force it directly into the building through distributing ducts. This method is generally known as the plenum system. The pressure produced in the building causes a continuous exit of the air from the building, either through the natural openings, as is usually the case in factories and other large buildings, or through special vent openings provided for the purpose, as in public buildings. This effectually prevents the entrance of cold air from without.

DOUBLE NIAGARA CONOIDAL PULLEY DRIVEN FAN WITH INLET BOXES TO DRAW THROUGH HEATER

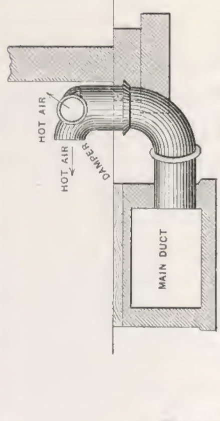
A second, and more common method for shop buildings where forced ventilation is not a necessity, is to draw the supply of air entirely from within the building and again force it through the distributing ducts, causing a continuous circulation of the air within the building. This often has an advantage over the plenum system in that all the heat supplied to the air is effective in heating the building.

An ideal arrangement is a combination of the plenum and return systems and this should be used wherever possible. By this method, the greater portion of the air is returned to the apparatus, but sufficient air is continuously taken from without through a fresh air





ELEVATIONS OF FAN



SECTION THROUGH HOT AIR DUCT

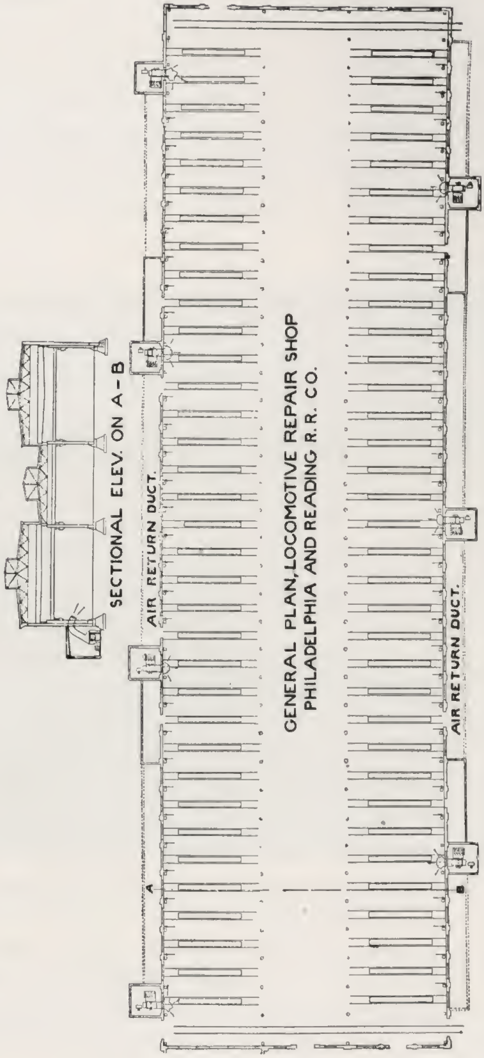
L. S. & M. S. RAILWAY MACHINE SHOP, COLLINWOOD, OHIO

connection to create a plenum within the building and prevent the inward leakage of the cold air along the floor. In this manner, the natural leakage is supplied—not by inflow of cold air through the crevices around the doors and windows—but by air passed through the apparatus and heated to an effective degree. This combination has been found by test to be more economical than air return alone. The proper amount of air to be introduced from without is determined by securing a point where the noticeable inward flow of air around the doors or windows ceases. If the plenum is carried beyond this point, there will be a loss due to unnecessary heating of the outdoor air.

LOCOMOTIVE MACHINE SHOP LEHIGH VALLEY R. R., SAYRE, PA.

Systems of Air Distribution

There are several systems of distributing the supply of heated air. A method usual in public and office buildings and sometimes employed in factory buildings, is the vertical duct system by which the air is admitted through vertical ducts or flues built in the walls and opening at a point about eight feet above the floor. Suitable openings are supplied at the floor line for the air that is forced out. By this method, the heated air is continuously forced downward as it cools and the cold air is always removed at the floor line. A method



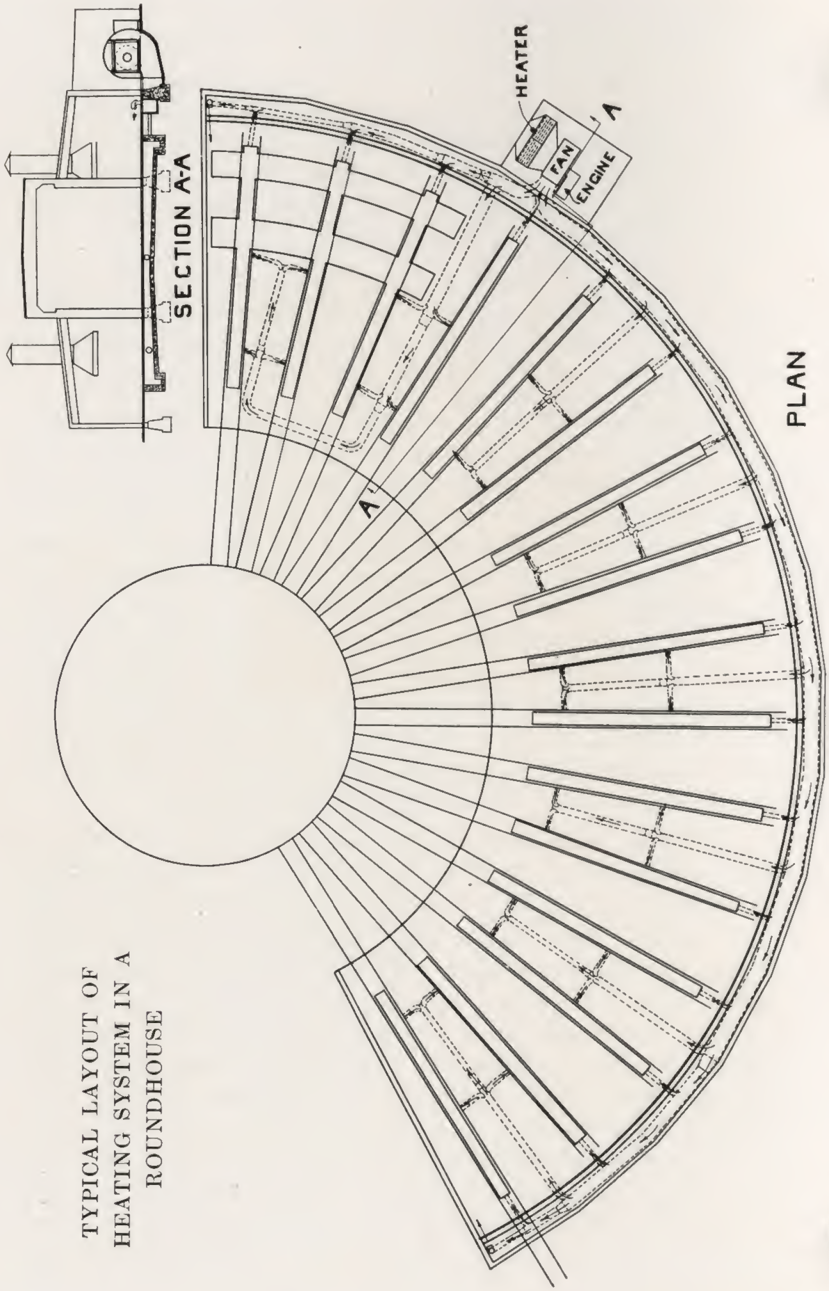
HEATING PLANT IN RAILWAY MACHINE SHOP, ILLUSTRATING SYSTEM OF DISTRIBUTED AIR RETURN, WITH UNDERGROUND AIR RETURN DUCTS

of distribution quite similar to this is one where the air is first blown into brick ducts placed underneath the floor. From these ducts vertical galvanized iron risers are arranged along the walls. These are arranged so as to blow downward and away from the wall at a height of about eight feet from the floor. These outlets should be adjustable so that, in case too direct a draft is caused in any portion of the building, the outlet can be turned in some other direction where the air current will not be objectionable.

This system is sometimes modified by placing the outlets close to the floor and blowing downward directly along the floor. This secures a perfect diffusion of the heated air at the floor line and avoids any draft which would be objectionable.

Excellent results can be secured by the use of overhead piping, providing it is not placed at too great a distance from the floor. The chief advantage of the overhead system is the saving in first cost, since on account of the high temperature and velocity of air in the distributing pipes, a great amount of heat can be transferred with a very small amount of material. The cost of the galvanized iron distributing system of air ducts is relatively small. The best results are secured with outlets at from 12 to 18 feet above the floor line. Above this height it is preferable to use drop pipes extending downward along the columns, where they will not interfere with traveling cranes. Such an arrangement of overhead piping is very frequently employed in foundries; while in large machine shops underground ducts are nearly always preferable.

Another system which has proved very satisfactory is that in which a distributing air return duct is employed. This approaches very closely in principle to the plenum system used in public buildings and is a combination of both plenum and exhaust systems. This may be best described by referring to the heating plant at the Philadelphia & Reading R. R. shops, at Reading, Pa. In this instance, several separate sets of apparatus have been provided, placed in small fan houses built at intervals at either side of the building. The peculiar feature in this installation is that no distributing ducts or piping for the heated air are used. The air is blown directly into the building at about 8 or 10 feet above the floor through an outlet branching in three directions. The distribution is effected entirely by the return vent ducts which are placed at frequent intervals along the walls. These open into large return air tunnels which are provided on either side of the building,



TYPICAL LAYOUT OF
HEATING SYSTEM IN A
ROUNDHOUSE

and serve the additional purpose of affording a convenient place for locating steam and water mains, and also electric light and power cables.

In many instances, an elaborate distribution is impracticable or undesirable. In such cases, a centrally located discharge pipe may be used. From this point the air is blown in all directions, and a circulation is produced by an exhaust connection to the fan inlet.

LEFT-HAND UP DISCHARGE, THREE-QUARTER HOUSING PLANOIDAL FAN, DIRECT-CONNECTED TO ENGINE

In such instances, very effective heating has been secured even where it was required to blow the air long distances. A good example of such a system may be found in the foundry building of the General Electric Company at Schenectady, N. Y. This foundry, which is probably one of the largest in the world, is heated in an entirely satisfactory manner with a few large branch outlets. Since the plant was installed, a large addition has been made. This new portion is heated by a branch outlet situated over 200 feet from the end of the original which shows how thorough distribution may be secured by forced circulation.

A typical installation of the Buffalo Fan System in a group of scattered buildings is in operation at the works of the Warren Featherbone Company, Three Oaks, Michigan, and is remarkable chiefly for the distance to which the heated air is transmitted from the apparatus to the various buildings.

BETHLEHEM STEEL CO., BETHLEHEM, PA.
ONE OF TWO UNITS INSTALLED IN NO. 3 MACHINE SHOP

The hot air piping for this building is carried entirely out of doors, and is protected by a wood boxing filled with sawdust. The loss of temperature, in passing through this piping, is determined by test to be only about 5°, which is remarkably good, considering the length and exposure of the piping. The entire system has given most satisfactory results, and possesses the important advantage of a central location of apparatus near the power house, thus utilizing the exhaust steam without long and expensive steam piping, and minimizing the amount of attention required.

Advantages of Fan System

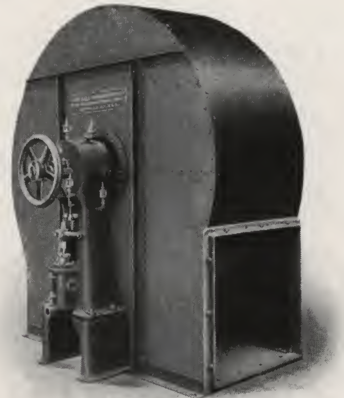
It has been shown that a most important source of economy with the fan system lies in the ability to secure a perfect distribution and diffusion of heat and by the production of a plenum preventing the cold air from entering the building and settling along the floors. Besides this the temperature is much more easily regulated with the Buffalo Fan System with separately controlled heater sections than with direct radiation, and thus a great loss which frequently occurs due to overheating, is prevented.

Utilization of Waste Heat

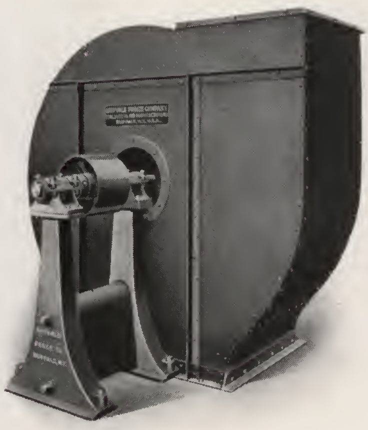
Another important point in economy is the utilization of waste sources of heat. By far the most common form of waste heat is from steam engines and other steam driven machinery. The ordinary simple engine running non-condensing has a water rate of about 32 pounds per H. P. and uses only 20% of the total heat of steam in work radiation, leaving a remainder of 80% available for the use in heating apparatus, which would otherwise be wasted. As the mean effective pressure in the ordinary engine cylinder may be placed at 40 pounds per square inch, an increase of one pound per square inch in back pressure reduces the effective H. P. of the engine 2½% and correspondingly increases the cost of the power production. In a compound engine the effect of back pressure is still more noticeable since the mean effective pressure, referred to the low pressure cylinder, may be placed at about 30 pounds per square inch; each pound of back pressure therefore reduces the power of the engine 3⅓%. It is, therefore, evidently unprofitable to use a system which will greatly increase the back pressure of the engine. The ordinary system of direct radiation used in shop buildings usually cannot be operated successfully without placing a back pressure upon the engine which is prohibitory. On the other hand the fan system heater is designed to circulate steam at very low pressure and can be operated successfully with ½ pound back pressure on the engine.

TWO SPECIAL NIAGARA CONOIDAL FANS USED IN CONNECTION WITH THE BUFFALO FAN SYSTEM OF HEATING AND VENTILATING, IN LARGE MANUFACTURING PLANT

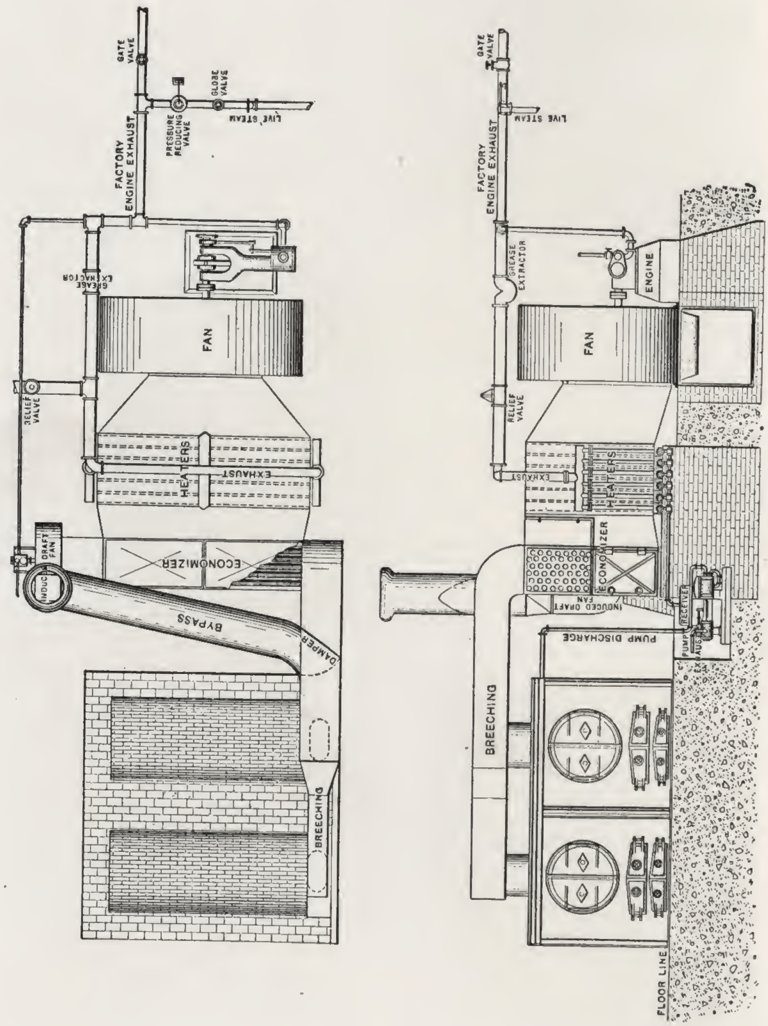
30″ to 60″ PLANOIDAL FAN WITH
OVERHUNG BLAST WHEEL



LEFT-HAND BOTTOM HORIZONTAL
DISCHARGE PLANOIDAL FAN
WITH VERTICAL CYLINDER
BELOW SHAFT ENGINE



70″ to 140″ PLANOIDAL FAN WITH
OVERHUNG BLAST WHEEL



IDEAL LAYOUT OF AIR ECONOMIZER

Air Economizer

The Buffalo Air Economizer is used to great advantage in plants where a large amount of steam is used in a manufacturing process. The layout on page 50 shows a typical plant using the Economizer. The building is heated by the Buffalo Fan System in connection with the Buffalo Air Economizer and the Buffalo System of Mechanical Induced Draft. With this system practically all of the exhaust steam can be used, and the economy and heating capacity of the boilers is materially increased. Sufficient air must be taken from without to take the place of air removed and to keep the building warm, otherwise condensation will occur.

Heating with Exhaust Steam

It is sometimes questioned whether it is cheaper to run an engine non-condensing and use exhaust steam for heating, or to operate it condensing and use live steam for heating purposes. The water rate of a compound Corliss engine at full load is about 20 pounds per H. P. non-condensing, and 14 pounds condensing, so that the water rate is 30% less when running condensing, than when non-condensing. The amount of heat available in the exhaust steam when running non-condensing is about 80%. Hence we see that the saving of steam running condensing is only 6 pounds per H. P., while the heat available in the exhaust steam is 16 pounds per H. P. and therefore a saving of 10 pounds of steam per H. P. could be made by operating non-condensing and using the steam in the heater if all the steam available could be utilized. We also see that there would be a saving as long as more than 38% of exhaust steam was utilized in the heater. With less economical engines, the saving made by running non-condensing and utilizing the exhaust steam is even greater.

With the steam turbine the water rate increases very much more rapidly with the increase of vacuum than with a steam engine. A steam turbine which with 28″ of vacuum has a water rate of 20 pounds of steam per K. W. hour at full load when running non-condensing requires 50 pounds of steam per K. W. hour at full load. Hence the use of exhaust from turbines without a vacuum is economical when the heating requirements are more than 60% of the steam consumption of the turbine running non-condensing.

Other sources of waste heat have been utilized to great advantage by means of the Buffalo Air Economizer in connection with the Buffalo Fan Systems of Heating and Mechanical Draft and the waste gases from the boilers, burning kilns, gas engines, etc. The

heat of these gases is being successfully used in many places for both heating and drying purposes. By this system it is possible to reduce the temperature of the boiler flue gases from 550° to 250°, thereby increasing the heating capacity and economy of the boilers approximately 15%. The saving effected by the utilization of these sources of waste heat frequently pays for the cost of installation in one season's operation.

NIAGARA CONOIDAL FAN DIRECT CONNECTED TO DOUBLE VERTICAL, DOUBLE-ACTING ENGINE

Flexibility of Operation The Buffalo Fan System possesses a great advantage over direct radiation systems in its flexibility of operation. With direct radiation a building heats up very slowly, and it is usually necessary to maintain a normal temperature all night in order to have it sufficiently warm in the morning. On the other hand the fan system with the proper amount of reserve can heat a building up in a short time. This allows the building to be cooled down during the night to just above freezing point, say an average temperature of 35° or 40°.

First Cost Besides these distinct advantages in economy over direct radiation there is usually a considerable advantage in first cost in favor of the Buffalo Fan System. This is due in part to the compactness of the system, requiring fewer connections and shorter lengths of steam mains, but more particularly to the great saving in amount

of radiating surface required owing to its greater effectiveness in the fan system. A determining factor in the rate of heat transmission of any heating surface is the velocity of air over that surface. This is shown by the curve on page 54 exhibiting the relation between air velocities and heat transmission as determined by accurate tests on the Buffalo Fan System heater. In direct radiation the heat is transmitted by convection currents and radiation only, while with the fan system an air velocity over the coils of from 1,000 to 1,200 feet per minute is usual; the former transmits only from 2 to 2.6 British Thermal Units per square foot per hour, per degree difference in temperature, while the fan system heater as shown by the curve on page 54, transmits from 10.4 to 11.5 B.T.U. per square foot per hour, per degree difference in temperature or about five times as much as direct radiation. Hence a correspondingly smaller amount of radiating surface may be used, which more than offsets the additional cost of fan, engine, and hot air piping.

The chief points of superiority of the Buffalo Fan System may be summarized as follows:

1. Perfect ventilation regardless of exterior conditions.

2. Uniform and proper distribution of heat.

3. High efficiency of heating surface (three to five times that of direct radiation).

4. Greatest economy in operation.

5. Utilization of exhaust steam.

6. Prevention of cold drafts from without by production of a plenum.

7. Independent regulation of heating and ventilating effects.

8. Great flexibility in operation to suit varying conditions, affording a maximum economy.

9. Ease of control, which prevents overheating.

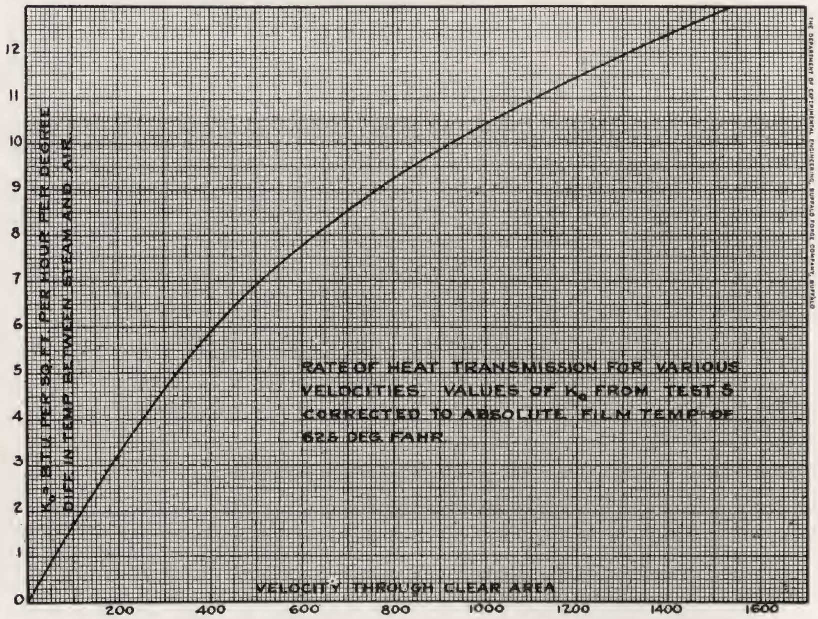
10. Great compactness, affording an economy of space and reducing the cost of steam connections.

11. Perfect drainage, making less repairs necessary and giving a lower rate of deterioration than with direct radiation.

12. Lower cost of installation.

13. The entire apparatus is easily portable and is, therefore, a permanent asset.

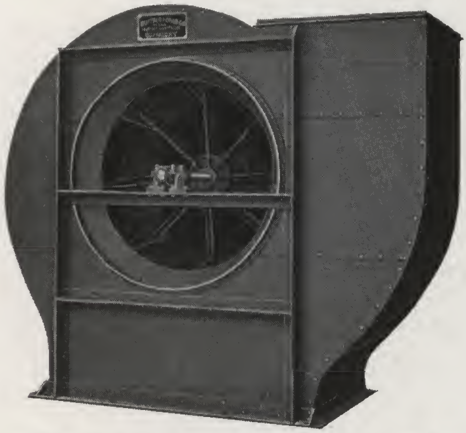
Round-houses The application of the Buffalo Fan System is particularly advantageous in the heating and ventilation of locomotive round-houses. These are especially difficult to heat on account of the large volume of warm air carried off through the open smokejacks which act as ventilators. A great deal of heat is absorbed, too, in the melting of the snow and ice on the locomotives and in the evaporation of the moisture thus produced. Ample ventilation is required to remove the smoke and steam produced by the engine and this necessarily consumes much heat. The air is drawn directly from out of doors and after passing through the coils of the heaters, is distributed by a system of underground ducts to the different stalls where it is discharged into the pits directly underneath the engine. Often the outlets in the pits are provided with adjustable elbows and dampers so that the blast of the hot air can be directed against any desired part of the engine or closed off entirely. More frequently, however, outlets are allowed to remain open at all times. By blowing the hot air directly underneath the engine, the snow and ice are melted in the shortest possible time and the moisture is



absorbed by the hot, dry air with great avidity. The distribution of the heat at the floor line places it where needed and permits it to be utilized to the fullest extent before the air passes out of the building. As the air is taken entirely from without doors, the necessary ventilation is secured at all times and a plenum is produced within, which tends to counteract the cold drafts occasioned by the frequent opening of the doors.

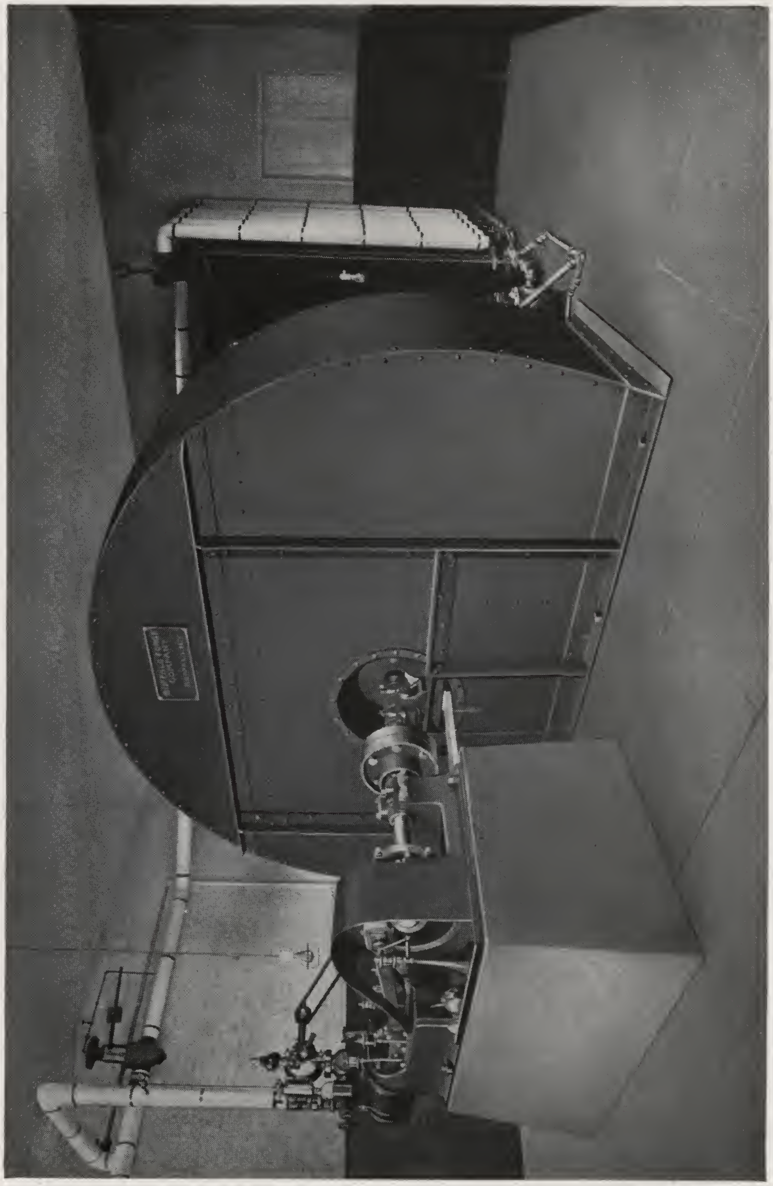
Application to Bonded Warehouses

The application of the fan system to bonded warehouses effects a great saving of time and money, and greatly improves the quality of the whiskey. In the old time bonded warehouse where there is no provision made for heating, the stock gains in proof dur-



STEEL PLATE PLANOIDAL TYPE "L" FAN

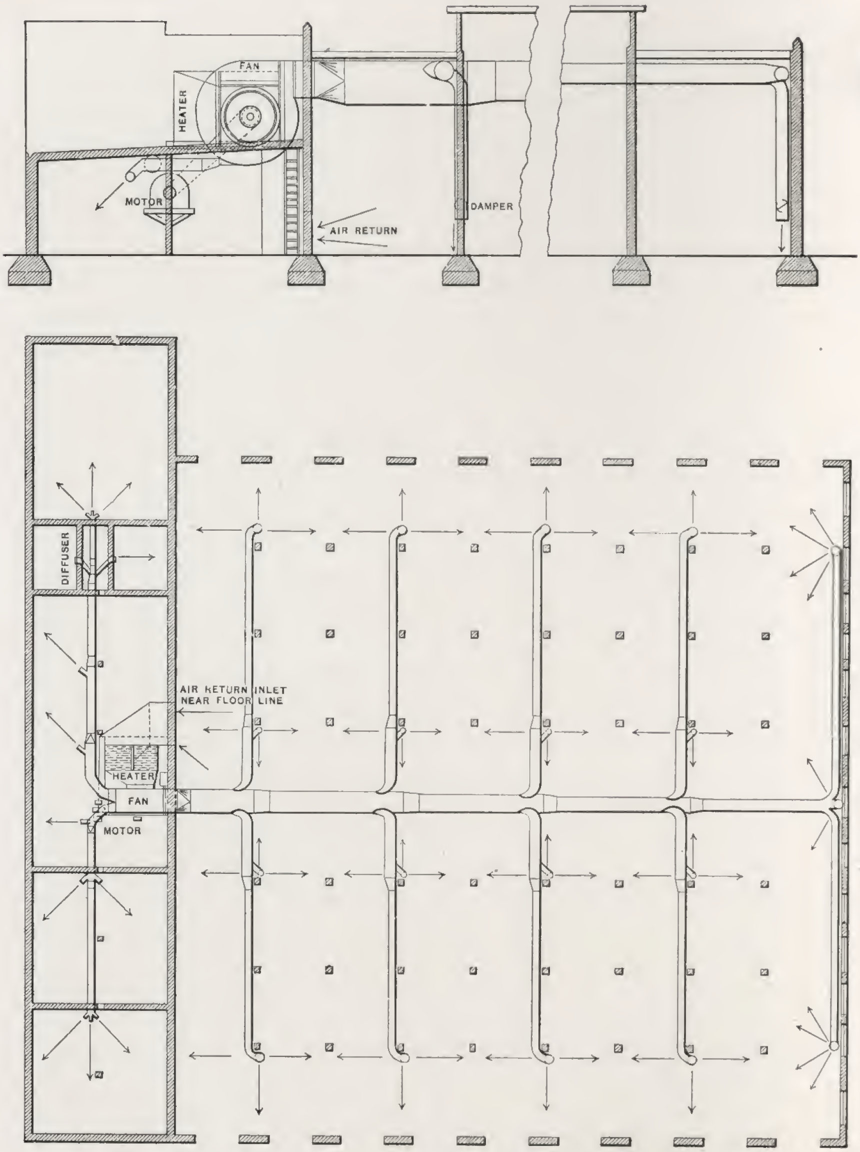
ing only four months of the year; during the remainder of the year, there is but little or no gain, and it is stored at a loss. The application of the Buffalo Fan System affords summer conditions as to temperature and ventilaton the year around, so that it requires but one-half to one-third of the time to gain the same proof and quality with the fan system as is required where no heating is provided. The two important results to be obtained by storage are increase of proof and removal of the volatile fusel-oil. To effect this, two conditions are essential: first, a constant temperature of from 75° to 80°, and second, a free and ample circulation of the air through all the required storage space. The latter condition, as the distiller



APPARATUS INSTALLED IN BONDED WAREHOUSE OF WATHEN DISTILLING CO., LOUISVILLE, KY.

well knows, is even more important than maintaining the temperature. This is due to the fact that both increase in proof and removal of fusel-oil are evaporating processes, in which a rapid air change is essential. Every distiller knows that in the old system with window ventilation, the barrels at the outside improve very much more rapidly than those at the center, whether the building is heated or not. This necessitates a frequent shifting of the barrels from the center to the outside and outside to center to secure uniform aging. With the fan system, however, warmed air is introduced directly underneath the floor throughout the entire storage space. This warm air, rising rapidly, gives all parts of the warehouse an equal ventilation and secures an absolutely uniform and rapid aging throughout; thus, the stock improves as rapidly during the cold weather as in the summer. Direct radiation, while it maintains the temperature, does not afford the ventilation obtained by the Buffalo Fan System, and on this account alone, the Fan System is far superior. Another very important advantage is that the apparatus can be placed entirely without the building, or at least, in a separate part of the building, thus making it unnecessary to enter the warehouse in order to regulate the temperature or care for the apparatus. The air ducts are so arranged as to comply strictly with the Government requirements. Whenever the warehouse is located near the distillery, as is usual, exhaust steam is always available in sufficient quantities for all purposes of heating. In such a case, the only cost of heating the warehouse is the cost of live steam required to operate the fan engine, which is a small item.

A very large number of Buffalo fan installations have been installed in bonded warehouses with marked success. As an illustration, we refer to the installation in the bonded warehouse of Freiburg & Workum's plant, Lynchburg, Ohio. This warehouse has a capacity of 45,240 barrels. The owners estimate that they are able to produce the highest grade of whiskey, both in quality and proof, in from one-third to one-half the time that was required before our system was installed, or, that in the same length of time, they are able to secure a correspondingly better quality than before. The cost of extra handling to change the position of the barrels on the rack may be entirely avoided with the fan system, thus a saving is effected which will more than offset the cost of operating the apparatus.



HEATING PLANT FOR PAINT SHOP, M. K. & T. R. R. CO., SEDALIA, MO., SHOWING COMPLETE AIR DISTRIBUTION FOR AVOIDING DRAFTS

Paint Shops

In paint shops it is desirable to dry paint rapidly and it is necessary to avoid drafts which agitate the dust and blow it about the building. With the fan system the former results are obtained by the introduction of dry air from without, and the latter is avoided by the use of unusually low air velocities and special arrangement of ducts. In the Buffalo system of paint shop heating, a combined plenum and exhaust system is frequently employed with most gratifying results. An illustration of this system applied to car paint shops is shown on page 58. The air is discharged through an overhead system at low velocities. A downward circulation is produced and all cold or moist air is removed at the floor line by exhausting a portion of the air through underground ducts opening into the pits under the cars. This system avoids all disturbing air currents and affords a perfect distribution of the heated air. In locations where a great deal of smoke and dust prevail, the Buffalo System of Air Purifying may be used to advantage. The rapidity of drying secured by the fan system far exceeds that obtained by any other method, owing to the frequent renewal of the air and its consequent greater drying effect.

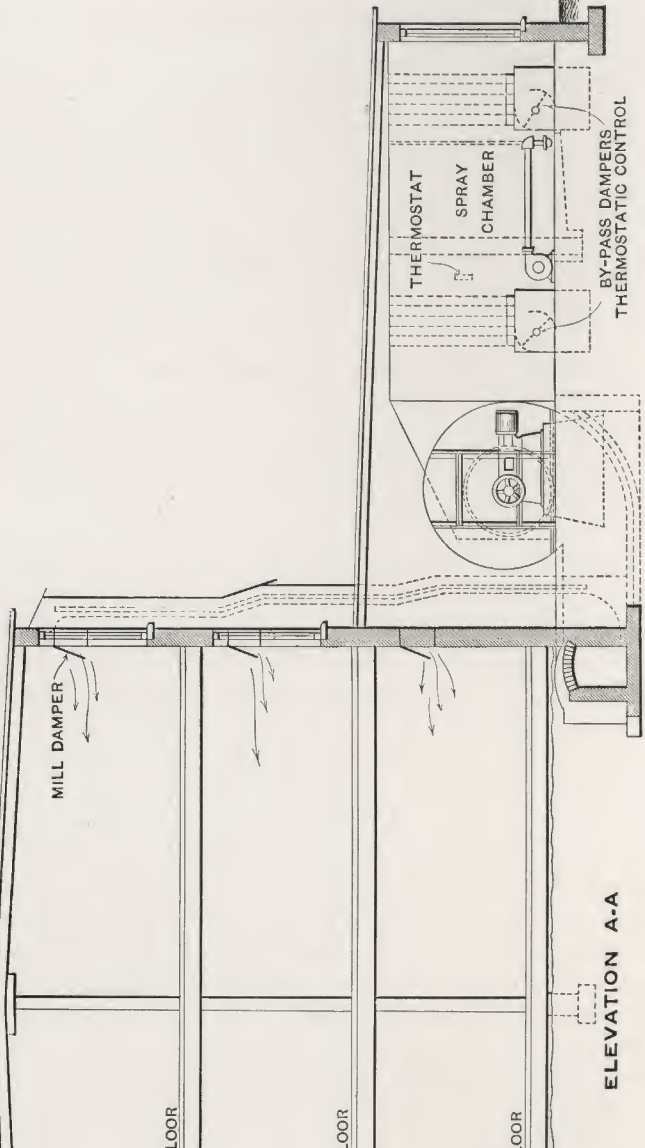
Paper Mills

In cold weather great trouble is ordinarily experienced in paper mills from the condensation produced from the moisture laden air coming in contact with the cold roof and walls. This condensation not only drops back on the dry paper producing blisters and thus injuring the product, but causes the roof boards and timbers to rot out very quickly. The most practical and satisfactory method yet devised is to blow hot air into the building just over the machines. A typical arrangement of Buffalo apparatus is to blow heated air against the roof and walls by a set of outlets, while another set of outlets is directly discharged against the machines. The first set of outlets keeps the roof warm while the air from the second set diffuses the steam remaining away from the machines and dissipates it. Air supplied is always drawn from without doors and exit for the moisture laden air is provided by louvres or ventilators in the roof. This insures a rapid absorption and removal of the moisture.

Textile Mills

In textile mills there is the additional problem of securing proper humidity together with ventilation.

Operators of textile mills have long appreciated the importance of correct humidity and temperature conditions in the spinning and



BUFFALO FAN SYSTEM HUMIDIFYING, HEATING AND VENTILATING COTTON MILLS

weaving processes. While these requirements were well understood, no entirely satisfactory or adequate method has heretofore been introduced for securing the desired results. These conditions which have such an important bearing upon the textile processes are:

First: the humidity, which is naturally quite insufficient for the best results during the greater part of the year, especially in the cold weather of winter and in the hot dry weather of summer. Second: the temperature, which should be maintained at from 70° to 75°, requires special heating in winter and cooling, if possible, in summer when the high outside temperature augmented by the weaving and spinning machinery becomes a great detriment. Third: ventilation, which, while not so important as the others commercially, is imperatively demanded from a humanitarian point of view, where so many women and children are required to work in a comparatively small space.

Humidity Control

In the manufacture of cotton goods it is very important that the proper humidity be maintained throughout the entire factory in order to get perfect thread and cloth. Correct humidity is also just as important in many other industries. Many methods, from the old fashioned watering can to elaborate and impractical designs, have been used to obtain desired results. Several of these systems neglected or entirely overlooked the fact that heating and ventilating are just as important as humidifying.

The most satisfactory and practical design of humidifying apparatus on the market at the present time is the Carrier System manufactured by the Carrier Air Conditioning Company of America, 39 Cortlandt St., New York City, and used in connection with the Buffalo Fan System of Heating and Ventilating. This system consists of five essential elements, the tempering coils, the humidifier, the heater, the fan, and the air distributing ducts.

The general arrangement of the apparatus is shown on page 60. The air is first drawn through a series of tempering coils controlled by the proper temperature for humidifying; thence it is drawn by the fan through the humidifier and forced through heater coils and by-pass where sufficient heat is imparted to it to maintain the desired temperatures in the rooms. By this arrangement the control of humidity is absolute and may be varied at will between any

desired limits. The mechanism is exceedingly simple in operation and relatively inexpensive. The temperature in the room is placed under absolute control without affecting the volume of ventilation. A uniformity of temperature and humidity is maintained.

When the air is taken from out of doors it is washed and purified as well as humidified. In this way fresh air is constantly supplied, enabling the operatives to work in a pure and healthful atmosphere, under all conditions of the weather.

Comparative Costs of Heating, Ventilating, and Humidifying

The question often arises as to the relative cost of heating, ventilating, and humidifying. As an example, assume a fan system of heating in a schoolroom, where outside temperature is 0° and room temperature is to be kept at 70°. Air must be raised to 70° before any heating will be done by it, therefore consider this amount of heat added for ventilation purposes.

The temperature of the air has to be raised still further for heating the room, and it is ordinarily assumed that air entering a room at 120°, with outside temperature 0°, will properly take care of heating requirements, and also furnish sufficiently rapid air change.

Accordingly 70° of 120° total or 58%, is used for ventilation and 42% for heating; and approximately the cost of ventilation is 60% and the cost of heating 40%, where humidifying is not considered.

Assuming that this same proportion holds for other temperatures, when the outside air is 40° and the room is to be kept at 70°, 30° or 58% is the amount of heating required for ventilation, and 22° or 42% for heating; and temperature of air entering the room should be 92°.

The amount of moisture which air will contain depends on its temperature. The amount of moisture actually contained at any temperature is called the absolute humidity; and the ratio of moisture which air actually contains at any temperature compared to what it could hold at that same temperature, is called the relative humidity. Thus, if a cubic foot of air contains 0.5 gr. of moisture at 0°, this being its absolute humidity, the absolute humidity will be 0.5 gr. when the air is heated to 70°. But a cubic foot of air at 70° would be capable of containing 8.0 gr. of moisture, therefore its relative humidity at 70° would be only about 6%.

When outside air is about 30°, it is well to have about 4 to 5.5 gr. of moisture per cubic foot of air, when temperature is raised to 70°; but with outside temperature 0° it is ordinarily considered that relative humidity should be about one-half the difference between outside and indoor temperatures, or where outside air is 0° and room temperature 70°, the relative humidity should be about 35%. This is a practical value which will not cause steaming of windows.

Assume in the above example that 35% relative humidity at 70° is to be maintained. The air then would leave the humidifier completely saturated at 41°, containing 2.85 gr. of moisture per cubic foot, and then could be raised to any desired temperature by passing over heating coils. As air entered at 0° containing 0.5 gr. of moisture per cubic foot, 2.35 gr. of moisture should be added to each cubic foot of air. Through the ordinary range of temperatures the absorption of one grain of moisture per cubic foot lowers the dry bulb temperature 8.5°, or 8.5° are necessary to raise moisture in a cubic foot of air 1 gr. or 20° will be necessary to raise moisture per cubic foot 2.35 gr.

This will be in addition to the 120° for heating and ventilating, or 140° will be required for heating, ventilating, and humidifying. Therefore 70° of 140° total, or 50%, is required for ventilating, 36% for heating, and 14% for humidifying, and it can be stated approximately that cost of ventilating will be 50%, cost of heating 35%, and cost of humidifying 15%.

COWL VENTILATOR INSTALLED AT LAFAYETTE HOTEL, BUFFALO, N. Y

FOUNDRY BUILDING, INTERNATIONAL HARVESTER COMPANY, SPRINGFIELD, OHIO

PART THREE

BUFFALO FAN SYSTEM APPARATUS

Arrangement of Apparatus

The Buffalo Fan System apparatus may be arranged either to exhaust the air through or to blow through the heater. Each arrangement possesses its own peculiar advantages but the selection in either case depends largely upon the individual requirements of the location. The exhaust-through apparatus possesses the advantage of greater compactness and a more convenient arrangement. On the other hand the blow-through apparatus is longer, but occupies a much narrower space. The former requires the use of an exhaust fan, which having but one inlet, is slightly less effective than the blower having two inlets, in the blow-through type; however, the exhauster discharges directly into the system of piping so that all the energy due to the velocity of discharge is utilized, while the blow-through system requires the change from a relatively high velocity at the fan outlet to a low velocity through the heater and back again to the higher velocity in the air ducts, with a consequent and unavoidable loss in pressure.

The exhaust-through apparatus is customarily employed in factory buildings on account of its compactness as well as the convenience in connecting directly with the piping system. The blow-through apparatus is necessarily used in public buildings and elsewhere wherever independent temperature regulation is demanded, as the use of a by-pass permits the independent distribution of hot air and cold air in any desired proportions.

The regular discharges of fans and blowers are designated as top or bottom horizontal discharge, up or down blast, and special, which are described by giving the angle of the discharge from the horizontal. The hand of a fan or blower is determined by the side on which the pulley or engine is located. Standing facing or nearest to the discharge outlet, the fan is right or left hand according to whether the pulley is on the right or left hand side.

Cone Fans

For exhaust ventilation and occasionally for forcing cold or tempered air into heating chambers, special locations may render the cone wheel suitable. This is a special form of fan wheel used without a housing, and is not at all to be compared with the disk or propeller fan, as it is distinctly of the centrifugal type.

Steel Plate and Multi-vane Fans

The Planoidal Type "L" steel plate fan is the result of extensive experimenting by our engineers and is a distinct improvement over the old steel plate type. This fan is now furnished with an inlet cone, and new proportions of wheels, housings, etc. were determined which materially increased capacity and reduced power consumption. We build double width steel plate exhausters with two inlets in this design, and capacity of such an outfit may be determined by doubling ratings given on pages 122 and 123.

The Niagara Conoidal Type "N" multiblade fan derives its name from the prevalence of conical shapes in its design. The inlet is conical, the blast wheel forms the frustrum of a cone, and the blades are curved over the tapering surface of a cone. All parts of this fan have been very carefully designed to give best efficiency under practical operating conditions. The wheels, blades and hub are designed so as to give the air a smooth easy flow without abrupt change of direction at any point. Also with this design, the back part of the blade cannot take up the greater part of the air which prevents uneven pressure and eddy currents and the air is distributed evenly over all parts of the blade.

Tests which we have made on various sizes of this fan show a very uniform velocity over the fan outlet and our standard guarantee is that velocity of air issuing from any part of the fan outlet as measured by a Pitot tube is not more than 15% above or below the average velocity. These fans are designed so as to make 100% of the outlet effective, and capacities which we give are for actual operating, not laboratory conditions. .

Niagara Conoidal fans are also made double width.

We have recently developed a new type of multiblade fan known as the Turbo-Conoidal shown on page 67. This is a special design of the Conoidal type and is especially adapted for direct connection to steam turbines or in any case where high speed is necessary. Tests on this type of fan have shown very high efficiency.

Selection of a Fan

It is well known and capable of demonstration in practice as well as in theory that of two straight blade fan wheels, the one having longer blades gives greater pressure, and that curving the blades forward in the direction of rotation increases the pressure, the converse also being true.

It is not a fact that a fan with forward curved blades is on that account any more efficient than one with radial blades; the two types have radically different characteristics, and each a field in which it excels; with the short forward curved blades good efficiency requires a greatly increased number as compared with the few blades of the radial type familiar in the steel plate fan.

In both types the need of careful design does not end with the proportions of the blades; the design of the scroll or housing, the area and position of the outlet and the diameter of the inlet are very important factors.

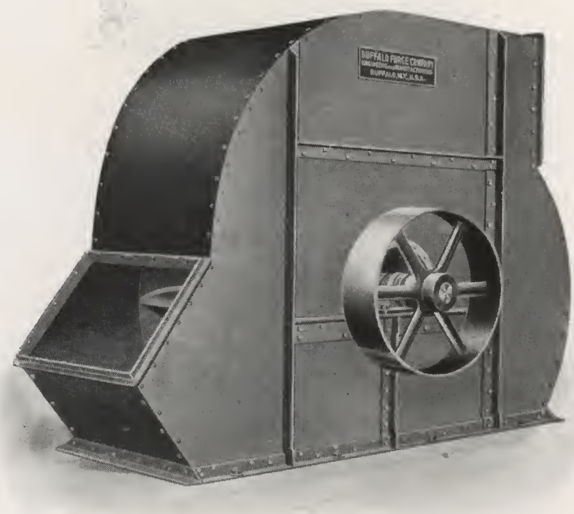
DOUBLE TURBO-CONOIDAL TYPE "T" FAN

With the older type, so long known as the steel plate fan or Planoidal, having 5 to 12 radial blades according to size, the pressure tends to build up as the capacity falls off, that is, at constant speed the pressure is considerably greater at half capacity than at normal rating. With the multiblade type, developing pressure by impact of the blades against the air rather than by centrifugal force, the pressure is greatest at normal load and decreases with capacities either above or below normal.

Thus we see in the case of a system where a uniform air quantity is desired, whether for heating, ventilating, forced draft or for drying processes the steel plate fan will come nearer giving this uniform quantity in spite of variations in resistance, throttling effect of closing dampers, and similar conditions.

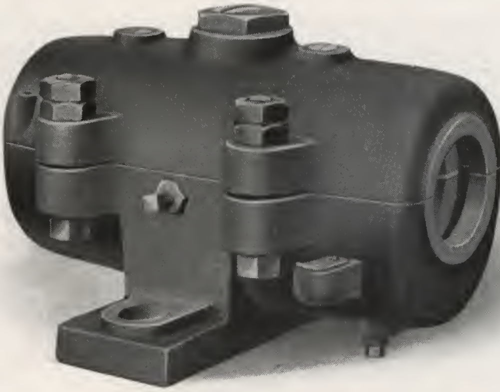
On the other hand, it is sometimes very desirable to be able to throttle the capacity of a fan without increasing the pressure and velocity, as for instance, if one wing of a building is closed off and it is not convenient to change the speed of the fan, the steel plate fan would deliver an increased amount of air into the remaining part of the system on account of its increased pressure, while the multiblade fan would be more sensitive to the increased resistance and would show only a slight increase in velocity through the ducts which remain open.

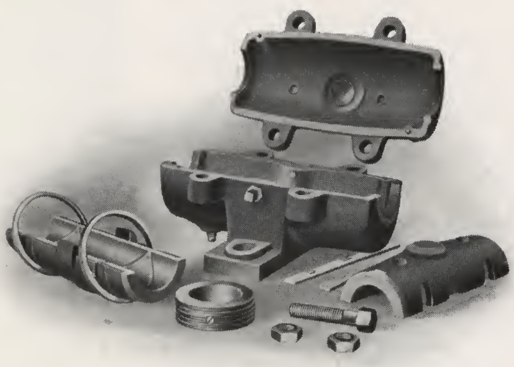
In general the multiblade fans, of which the Niagara Conoidal is a type, require less space than steel plate fans of equal capacity and efficiency, and frequently have an advantage on account of using higher operating speeds, but are much more sensitive to changes in resistance. For this reason when Niagara Conoidal fans are used instead of Planoidal fans it is necessary to use greater precautions in designing the duct systems, determining all the frictional resistances closely, and selecting the proper speed for the size of fan to be used.



DOUBLE DISCHARGE PLANOIDAL FAN, LEFT-HAND TOP HORIZONTAL AND RIGHT ANGULAR UP DISCHARGE, THREE-QUARTER HOUSING

BUFFALO SPHERICAL TYPE FAN BEARING



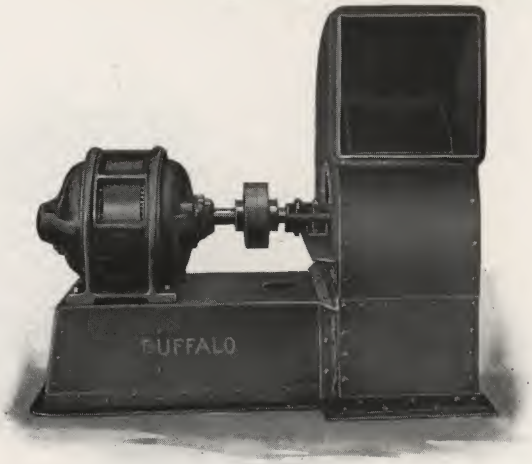


This dust proof and oil tight bearing consists of a split sleeve lined with babbitt and completely encased in the bearing housing. The sleeve is mounted between spherical surfaces which allow the bearing to adjust itself in every direction, and the housing provides a large oil reservoir in which two oil rings dip; overfilling is prevented by the position of the opening through which the oil is supplied and which also indicates the oil level.

In the interest of safety the thrust collar is placed inside the housing, running against a babbitted shoulder; grooves on the outside surface of the thrust collar throw off all oil and absolutely prevent it from creeping along the shaft and being drawn into the fan.

Motor Driven Fans

We consider it best in most cases to install engine driven ventilating fans, preferably direct-connected, as this method is more economical and permits wider speed variation. In many cases, the steam pressure necessary—15 to 20 pounds for low pressure engines,—is not available, or the location is such that apparatus requiring less attention must be used; here motor drive affords the solution, and special designs for electric fans have been made by this company, which combine the necessary strength and rigidity with a pleasing appearance. The motor bases are rigidly attached to the fan housings, and are of a heavy cast iron one-piece box construction, tapering to a broad base, and finished off with heavy angle iron. The base is stiffened across the interior with steel

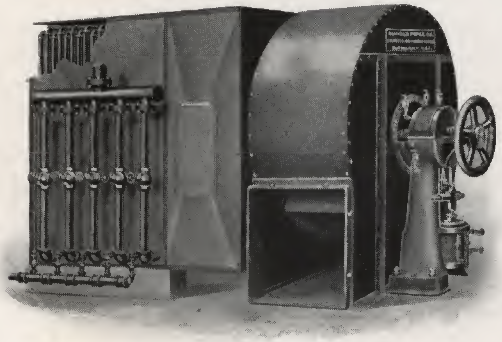


BUFFALO MOTOR DRIVEN PLANOIDAL STEEL PLATE FAN

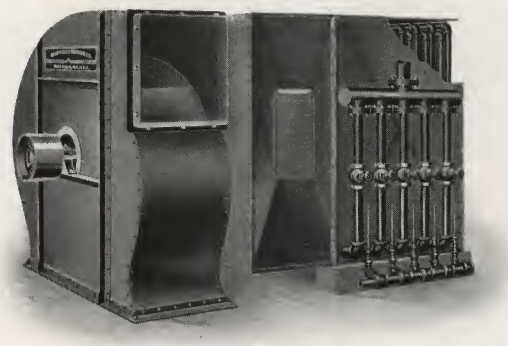
ribs and is made with corners rounded so as to avoid an unfinished appearance.

Motors may, in the case of exhaust fans with one inlet, have the fan wheel overhung on the motor shaft, which for this purpose is extended, but it is usually preferable to use a coupling and place a bearing on the side of the fan furthest from the motor. The requirements of alternating current motors make it impossible to use direct connected arrangement for heating and ventilating fans, except in rare cases, as the low speeds would require motors with a very large number of poles and a prohibitively high cost. For direct current, motors are readily manufactured for any desired speed,

and although a slow speed motor is considerably more expensive than a high speed motor of the same power, the advantage of the construction is sufficient to warrant its adoption except for the



RIGHT-HAND BOTTOM HORIZONTAL DISCHARGE PLANOIDAL STEEL PLATE FAN, DRAWING THROUGH HEATER



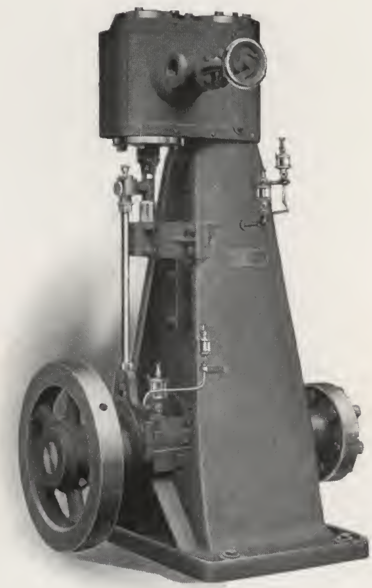
LEFT-HAND TOP HORIZONTAL DISCHARGE PLANOIDAL STEEL'PLATE PULLEY EXHAUST FAN, DRAWING THROUGH HEATER

largest sizes of ventilating fans which also operate at the slowest speeds. For these large sizes, excellent results have been had from the use of silent chain drives, although nothing of this kind can be considered better than a belt.

Engines

This company is very fortunately situated in having its own completely equipped engine department making no less than nine distinct types and the design as well as the construction materials

of its fan engines are as carefully watched as in the well known Buffalo Automatic Engines for electric light and power service, which are built in the same shops. The engines supplied with heating apparatus are in many cases identical except for variations in equipment and fittings.



CLASS "O" VERTICAL FAN ENGINE

The various types and the service for which they are adapted may be set forth clearly.

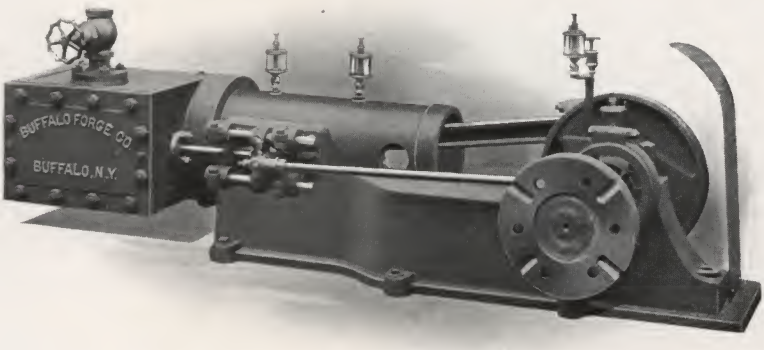
Class "A" and Class "B" horizontal center-crank and side-crank, built for high or low steam pressures, and largely used in schools and public buildings. Shown on pages 37 and 38.

Class "A" and Class "B" vertical engines are also built for high or low pressures. See page 35.

Class "I," inverted engines (cylinder below shaft). Very compact and simple. Designed for full housing fans only. Used with Planoidal Type "L" fans up to 150″ and with Niagara Conoidal Type "N" fans up to No. 11. The working parts are not exposed to injury, and size of the wearing surfaces is unusually large. A balanced piston valve insures much better steam economy than the average small engine permits.

Two designs used extensively with fans are shown on pages 72 and 73, the Class "O" vertical and the Class "N" horizontal tangye-frame side-crank. The aim has been to produce an engine free from delicate or complicated parts, and easily accessible, but with no essential feature omitted. The expense for finish is limited to the vital parts, yet both types have a pleasing effect of strength and careful design.

Other types less frequently used in heating and ventilating work are also built by this company; these include the vertical two cylinder double and single acting for high speed work, and the large side-crank engines of the box girder type, for mine ventilating and large mechanical draft fans.



CLASS "N" HORIZONTAL TANGYE-FRAME SIDE-CRANK ENGINE

The Buffalo Fan System Heater

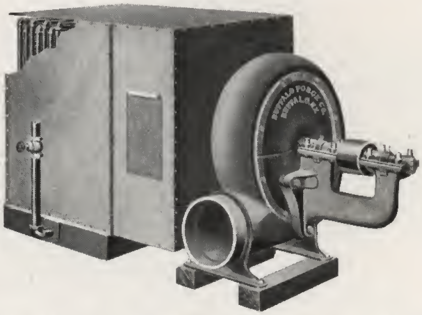
The Buffalo Fan System heater has been designed with due regard to peculiar requirements and to the end of securing a maximum effectiveness.

First: The heater is so arranged as to insure perfect drainage. The pipes are immediately relieved of all condensation and the design of the base allows no opportunity for pocketing of water, and thus avoids damage from freezing. The drain ports are made large to allow for an unusually rapid condensation without choking and filling. Water hammer is avoided.

Second: A perfect circulation of the steam is obtained by the progressive nature of the steam flow enforced, which removes all air from the coils, carrying it to a single chamber in the base whence it is readily removed. Air binding, the greatest enemy of radiating efficiency, is thus prevented.

Third: The air passages between the coils are so proportioned as to secure the highest efficiency of radiating surface compatible with a low resistance to air flow. One of the essentials to this end is to bring the air in intimate contact with all parts of the heating surface. A second and even more important requirement is to maintain a uniform and maximum velocity throughout. That the air velocity is a determining factor in the rate of heat transmission is conclusively shown by the data from heater tests given on page 54. By avoiding a fluctuation from high to low velocities, an unnecessary loss of pressure of the air in passing through the heater is prevented.

Fourth: Each section independently connected and controlled by valves so that steam may be admitted to as few or as many sections as desired, giving the operator a convenient and absolute



RIGHT-HAND BOTTOM HORIZONTAL DISCHARGE
"B" VOLUME EXHAUST FAN WITH
INDIRECT HEATER

control of air temperature and heating effect—an advantage which is easily appreciated. By this method of connection, any section may be removed for repairs in case of leakage without interfering with the operation of the remaining sections. This construction permits live steam to be introduced into any number of sections and exhaust steam to be used for the remaining, or live and exhaust steam may be used in the same section at the same time.

Heater Dimensions

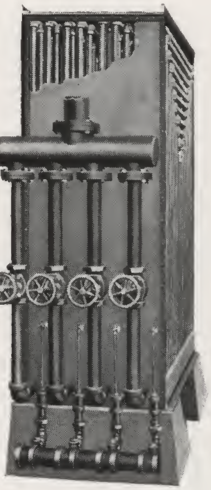
On pages 126 to 129 are given the tables of sizes of engines and sizes and capacities of heaters for various sizes of fans. All sections contain four rows of pipe. When an apparatus requires a clear area through the coils greater than that of the largest heater, two smaller coils are placed back to back as shown on page 7. When still greater area is required, a triplex arrangement in three groups is used.

BUFFALO FAN SYSTEM APPARATUS WITH "B" VOLUME EXHAUST FANS

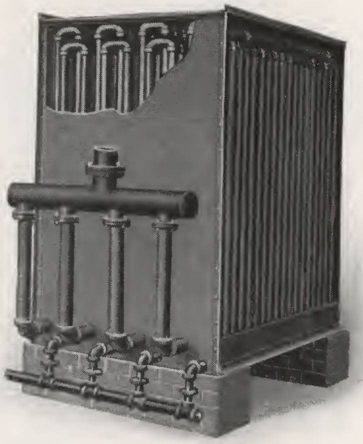
Size of Fan	Diameter Outlet in Inches	Diameter and Face of Pulley	Ordinary Speed of Fan	Cubic Feet Air per Minute	Total Lineal Feet of Heater	Size of Heater Sections	Type of Heater Sections	Number of Heater Sections	Floor Space of Fan and Heater	Weight of Fan and Heater in Lbs.
4B	9	5 x 3⅞	1216	828	354	8 x 8 x 40½	Indirect	2	6′–10″ x 2′– 6″	1250
5B	10⅝	5¾ x 4⅝	1100	1200	442	8 x 10 x 40½	Indirect	2	7′– 4″ x 2′–11″	1600
6B	11	6½ x 5¼	1275	2382	692	10 x 12 x 40½	Indirect	2	8′– 1″ x 3′– 4″	2450
7B	14	7½ x 6¼	1170	2750	692	10 x 12 x 40½	Indirect	2	8′– 5″ x 3′– 8″	2650
8B	16⅜	8½ x 7¼	1000	4000	928	12 x 14 x 40½	Indirect	2	9′– 3″ x 4′– 3″	3500
9B	17⅞	9½ x 8¼	825	6650	1075	3′ x 4′–4″	R. O. A. P.	5	9′– 5″ x 4′–10″	3900

BUFFALO FAN SYSTEM APPARATUS WITH PLANOIDAL STEEL PLATE EXHAUST FANS

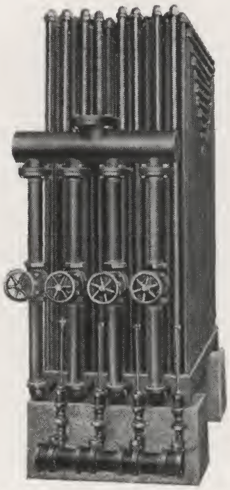
Size of Fan	Size of Outlet in Inches	Diameter and Face of Pulley	Ordinary Speed of Fan	Cubic Feet Air per Minute	Total Lineal Feet of Heater	Size of Heater Sections	Type of Heater Sections	Number of Heater Sections	Floor Space of Fan and Heater	Weight of Fan and Heater in Lbs.
30 in.	10½ x 10½	8 x 3	816	1350	442	8 x 10 x 40½	Indirect	2	6′–8″ x 2′– 2″	1600
35 in.	12¼ x 12¼	9 x 3	780	2000	552	10 x 10 x 40½	Indirect	2	7′–5″ x 2′– 7″	2050
40 in.	14 x 14	10 x 3	685	2700	692	10 x 12 x 40½	Indirect	2	7′–7″ x 2′–11″	2600
45 in.	15⅞ x 15⅞	11 x 3	650	3600	774	10 x 14 x 40½	Indirect	2	7′–9″ x 3′– 4″	2900
50 in.	17⅝ x 17⅝	12 x 4	650	5000	930	3′ x 3′–10″	R. O. A. P.	5	8′–2″ x 3′– 8″	3400
60 in.	21¼ x 21¼	16 x 5	580	7650	1215	3′ x 4′–10″	R. O. A. P.	5	8′–7″ x 4′– 4″	4200



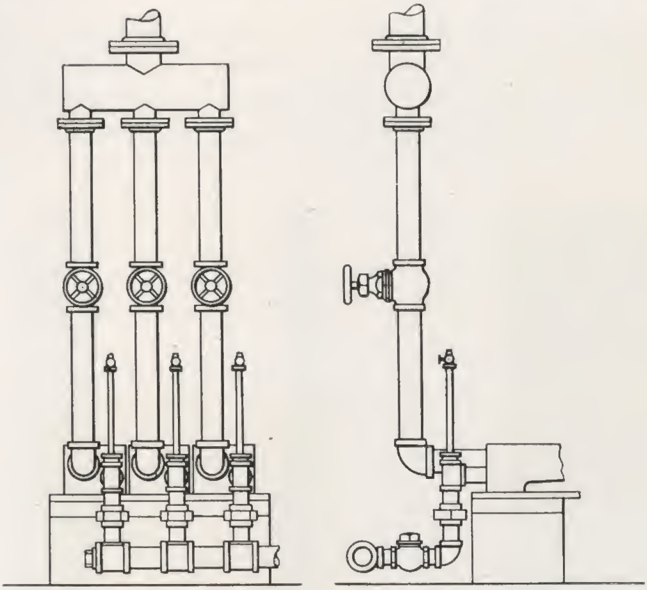
BUFFALO REGULAR
OPEN AREA PATTERN
HEATER WITH CASING



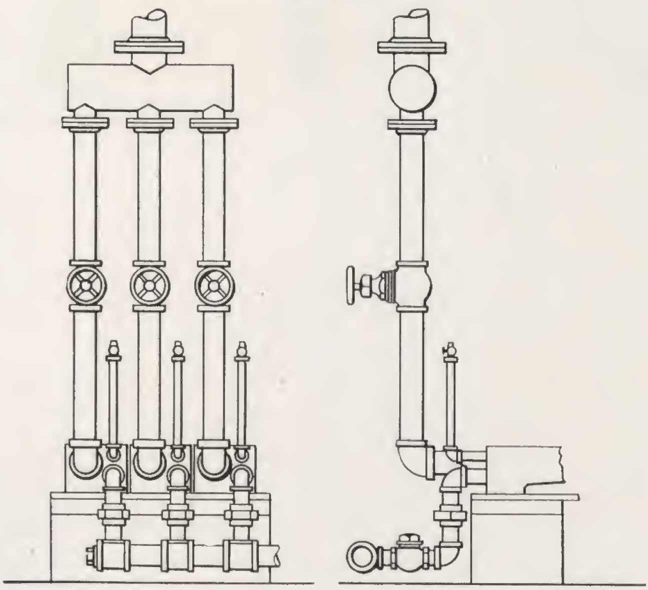
BUFFALO RETURN BEND HEATER



BUFFALO REGULAR
OPEN AREA PATTERN
HEATER WITHOUT
CASING



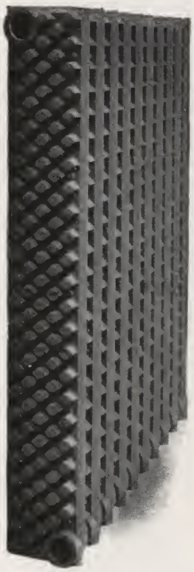
STEAM, DRIP AND AIR CONNECTIONS FOR REGULAR OPEN AREA PATTERN HEATERS



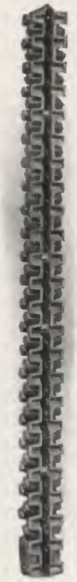
STEAM, DRIP AND AIR CONNECTIONS FOR RETURN BEND HEATERS

Vento Heaters

The Vento cast iron heater, an illustration of which is shown below, is designed especially for use in fan and blower work. These heaters are made in sections of various heights which may be so assembled as to make a heater of any desired size and depth.



REGULAR SECTION



NARROW SECTION

Indirect Heaters

In designing heating and ventilating equipments, it is sometimes desirable to locate the fan away from the building, either in the power house or specially built apparatus room. If the distance be considerable it is more economical to place the heating surface in the building itself, carrying the unheated air over the intervening space, than to first heat it. The condensation and heating capacity from a given amount of properly designed radiation, is from three to five times greater with a forced circulation of air than in ordinary plants. Obviously, the heater designed for a fan system, therefore, must provide for positive and unusually rapid condensation, in order that the coils may be invariably hot. The Buffalo Indirect Heaters are also used with Planoidal and Niagara Conoidal Fans and "B" Volume Blowers for small installations.

SIZES AND DIMENSIONS OF BUFFALO STANDARD HEATERS

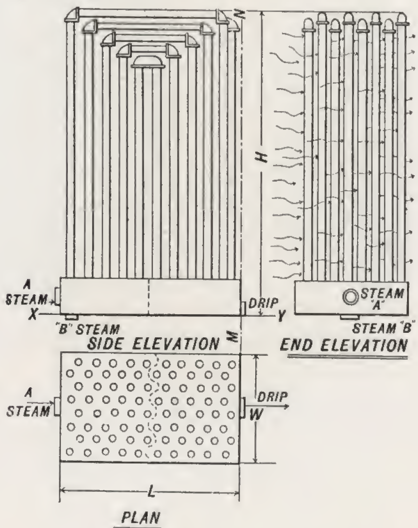
Manner of Piping	Number of Pipes	Length of Section	Section Number	Extreme Height Section	Width of Section	Lin. Feet of 1 Inch Pipe per Section	Total Effective Sq. Ft. Heating Surface	Equivalent in Lin. Feet of 1 Inch Pipe	Clear Area for Air Passage Sq. Ft.	Weight
R.O.A.	56	3′ 4 row	1 A	3′- 4″	8½″	140	54.7	159	4.4	473
			2 A	3′-10″	8½″	168	64.2	186	5.2	515
			3 A	4′- 4″	8½″	196	74.0	215	6.0	565
			4 A	4′-10″	8½″	224	83.7	243	6.8	616
			5 A	5′- 4″	8½″	252	93.3	271	7.6	656
			6 A	5′-10″	8½″	280	102.5	298	8.4	708
R.O.A.	72	4′ 4 row	1 B	5′- 4″	8½″	320	119.0	346	9.7	819
			2 B	5′-10″	8½″	356	131.5	382	10.7	877
			3 B	6′- 4″	8½″	392	143.9	418	11.2	938
			4 B	6′-10″	8½″	428	156.5	455	12.6	1003
R.O.A.	80	4′-6″ 4 row	1 C	5′-10″	8½″	396	148.2	431	12.1	997
			2 C	6′- 4″	8½″	436	162.0	480	13.1	1055
			3 C	6′-10″	8½″	476	174.8	507	14.2	1127
			4 C	7′- 4″	8½″	516	188.6	548	15.3	1174
R.O.A.	88	5′ 4 row	1 D	6′- 4″	8½″	476	174.3	507	14.1	1182
			2 D	6′-10″	8½″	520	189.3	550	15.4	1262
			3 D	7′- 4″	8½″	564	204.8	595	16.6	1325
			4 D	7′-10″	8½″	608	219.8	638	17.7	1407
R.O.A.	104	6′ 4 row	1 E	7′- 4″	8½″	674	245.0	712	19.8	1505
			2 E	7′-10″	8½″	726	262.9	763	21.3	1600
			3 E	8′- 4″	8½″	778	280.8	816	22.7	1695
			4 E	8′-10″	8½″	830	298.7	868	24.2	1770
R.O.A.	64	7′ 2 row	1 F	8′- 4″	6″	477	173.1	503	28.1	1198
			2 F	8′-10″	6″	509	184.3	535	30.0	1244
			3 F	9′- 4″	6″	541	195.3	567	31.7	1303
			4 F	9′-10″	6″	573	205.3	596	33.3	1350
R.B.	128	7′ 4 row	1 G	7′- 4″	8½″	796	291.0	845	23.6	1845
			2 G	7′-10″	8½″	860	313.2	910	25.4	1950
			3 G	8′- 4″	8½″	924	335.2	974	27.2	2055
			4 G	8′-10″	8½″	988	357.2	1037	29.0	2160
			5 G	9′- 4″	8½″	1052	379.2	1101	30.7	2280
			6 G	9′-10″	8½″	1116	401.2	1163	32.5	2380
R.B.	154	8′-6″ 4 row	1 H	8′- 4″	10″	1119	410.2	1190	33.2	2675
			2 H	8′-10″	10″	1196	436.8	1265	35.3	2800
			3 H	9′- 4″	10″	1273	463.5	1345	37.6	3075
			4 H	9′-10″	10″	1350	490.0	1421	39.8	3200
			5 H	10′- 4″	10″	1427	516.6	1499	41.8	3325
			6 H	10′-10″	10″	1504	543.2	1578	44.0	3455
R.B.	170	9′-6″ 4 row	1 I	8′- 4″	10″	1231	452.3	1313	36.7	3205
			2 I	8′-10″	10″	1316	481.6	1396	39.0	3350
			3 I	9′- 4″	10″	1401	510.9	1481	41.4	3485
			4 I	9′-10″	10″	1486	540.2	1570	43.8	3625
			5 I	10′- 4″	10″	1571	569.5	1651	46.0	3770
			6 I	10′-10″	10″	1656	598.7	1739	48.4	3910
			7 I	11′- 4″	10″	1741	628.0	1821	50.8	4060
			8 I	11′-10″	10″	1826	657.3	1910	53.2	4200

R.O.A.=Regular Open Area Pattern.

R.B.=Return Bend.

Indirect Heaters

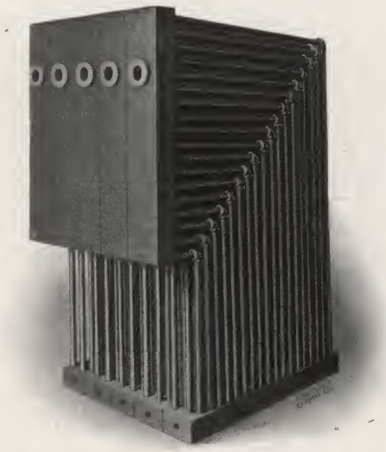
As the table and engravings show, a variety of sizes are built, the smallest being 6 pipes wide and 8 pipes long. Under the heading of "Size," the first row of figures gives the number of pipes across the steam supply and drip ends, and the second column the number of pipes in the length of the coil. Cast iron manifolds are used for the bases into which the pipes are screwed, as in the regular fan system heaters. The indirect heaters may be used in an upright or horizontal position, according to the requirements. These heaters are known as the solid base type, and a diaphragm in same compels the steam to flow evenly through all pipes. The steam supply enters the heater base at one end and the water of condensation is removed directly opposite. These coils are designed for the use of either live or exhaust steam, being effectively applicable for low pressure.



ACTUAL LINEAL FEET 1″ PIPE IN EACH SECTION

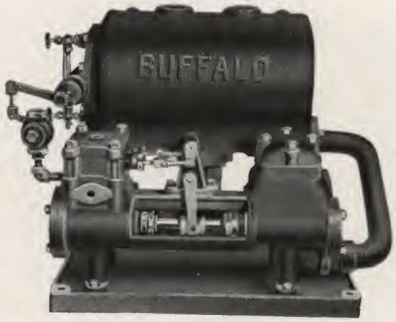
Size	40½″	46½″	52½″	58½″	64½″	W	L
6 x 8	133	154	177	198	221	12½	22
8 x 8	177	206	236	265	295	16¼	22
8 x 10	221	258	295	332	369	16¼	27
10 x 10	276	323	369	415	462	20	27
10 x 12	346	387	443	498	553	20	32
10 x 14	387	451	517	581	645	20	37
12 x 12	398	464	532	598	663	23¼	32
12 x 14	464	542	618	697	774	23¼	37
12 x 16	532	618	709	798	886	23¼	42
14 x 14	542	632	723	814	906	27½	37
16 x 16	708	827	945	1061	1181	30¼	42

BUFFALO FOUR-ROW OPEN AREA PATTERN SECTION



BUFFALO MITER-COIL HEATER, WITHOUT CASING OR CONNECTIONS

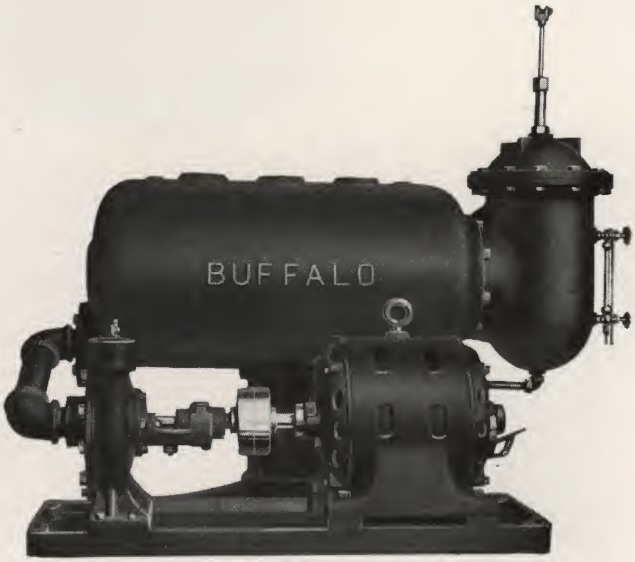
BUFFALO FOUR-ROW SECTIONAL MITER-COIL HEATER



DUPLEX AUTOMATIC FEED PUMP AND RECEIVER

Feed Pumps and Receivers

A Buffalo Pump and Receiver is used for returning condensation to the boilers. If steam pressure is above 30 to 50 pounds, depending on size of pump, an outfit as shown above is used. For lower steam pressures, a low steam pressure duplex pump and receiver, with cylinders properly proportioned, may be used. For steam pressures below 10 to 25 pounds, a centrifugal pump and receiver as shown below is recommended. These outfits are made automatic and will start and stop, depending on the rise and fall of water in the receiver.



CENTRIFUGAL AUTOMATIC FEED PUMP AND RECEIVER

Gas Heaters

The adaptation of the Buffalo Gas Heater to provide for warming the air entering a heating and ventilating system represents a field for use quite distinct from those employing steam heated radiating coils. Its use is applicable to any situation where economy and particularly cleanliness, minimum amount of apparatus, and automatic operation are desirable features. Reports of tests by Mr. Jay M. Whitman read before the A. S. M. E. in December, 1905, show efficiency of gas fired steam boilers to be seldom in excess of 65 to 70%—yet this is exceeded by far in the guaranteed efficiency of the Buffalo Gas Heater with the unique arrangement for the return of a portion of the flue gases. Besides the direct heat furnace is much cheaper to install than a gas fired boiler and steam coils—hence its wide application in natural gas belts or where fuel gas can be obtained at ordinary cost.

Two large Buffalo Gas Heaters are installed for heating the storage and warehouse building of the American Rolling Mills, Middletown, Ohio. This building is heated without the use of boilers, steam heater coils or direct radiation, the heat of the burning gas being transferred direct to the air which is to be distributed.

In appearance and design Buffalo Gas Heaters resemble a horizontal water tube boiler. Each heater consists of a bank of American ingot iron boiler tubes expanded at both ends into a heavy boiler plate. In these heaters six Gwynn gas burners are located in the front fire brick combustion chamber, and are so designed that abundant outside air is obtained for combustion. This chamber is so designed that complete combustion will occur in it, and thus insure the greatest possible amount of heat from the gas. The grates are so arranged that coal or oil may be used, if it ever becomes necessary to abandon the use of gas.

In the installation referred to, the hot gases are baffled to the back of the heater at a temperature of 3000° F. These are too hot to put directly into the tubes and this trouble is avoided by recirculating the gases emanating from the cold end of the horizontal tubes, and incidentally the efficiency is increased as waste heat up the stack is obviated. Two-thirds of these cold gases at approximately 400° F. are sucked back into the return air chamber by a double discharge Niagara Conoidal fan, and the hot gases of combustion at 3000° F. and the recirculated gases at 400° F. are here mixed, giving a resultant temperature of 1400° F. The gases then pass up and back through the inside of the horizontal tubes. The pure air for distribution in the building is drawn through the clear area around the tubes,

TWO VIEWS OF GAS HEATERS AT AMERICAN ROLLING MILLS,
MIDDLETOWN, OHIO

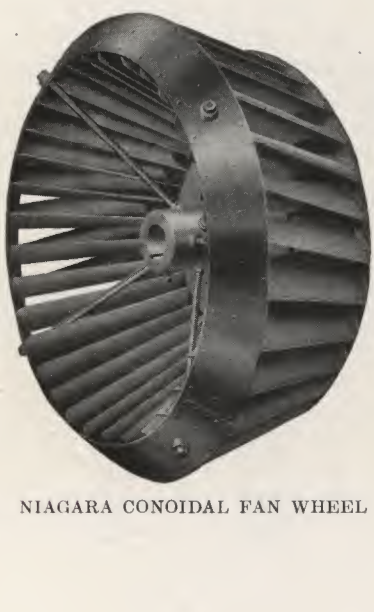
where the heat transfer occurs, and the heated air is forced through the duct system by a Niagara Conoidal Fan.

A complete and thorough test of this installation was made by our engineers, and the two heaters showed an average operating efficiency of 85% without considering radiation losses.

Taking the average heat value of coal at 11000 B. T. U. per pound and gas at 1000 B. T. U. per cubic foot, it is seen that one ton of coal has the same heat value as 22000 cubic feet of gas; but a boiler operates at only 50% efficiency, while the gas heater has an efficiency of 85%, therefore one ton of coal actually has the same heating value as 12900 cubic feet of gas. In many places this ratio makes it advisable to use gas heaters instead of steam boilers.

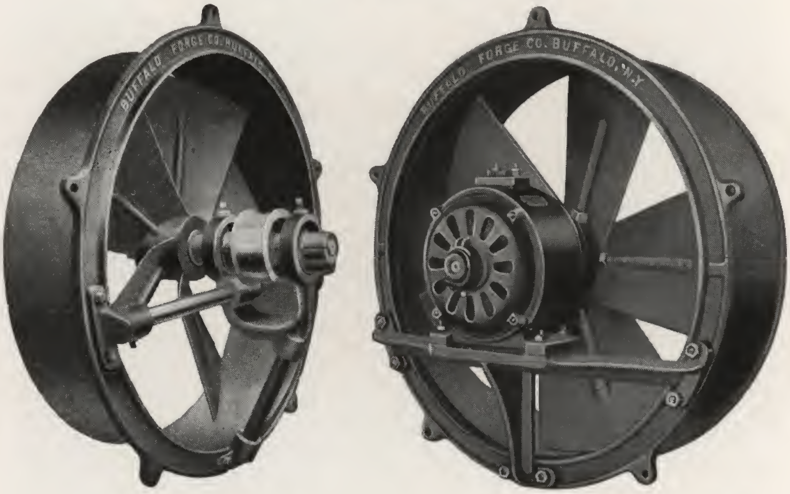
With Producer Gas

It is interesting to note that where natural gas is not available a gas furnace heating system may be operated quite as economically as a steam heating system. The average gas producer in the market, using soft coal, will give an efficiency of from 65 to 70%. This would give a combined efficiency of gas producer and gas heater of from 55 to 60%. In such a system it is customary to provide for the utilization of the exhaust from the gas engines. This exhaust alone is frequently sufficient to heat the entire building in moderate weather.



NIAGARA CONOIDAL FAN WHEEL

BUFFALO DISK FANS



All sizes of pulley-driven disk fans up to and including 48 inch are made with oil ring bearings arranged on the same side of the fan, which avoids the troublesome and dangerous practice of reaching through the fan blades when oiling. Larger sizes are made with plain bearings and an overhung pulley.

Motor driven disk fans up to and including 30 inch have motor secured to the frame of the disk wheel by a bracket. In large sizes, motor is supported by a tripod.

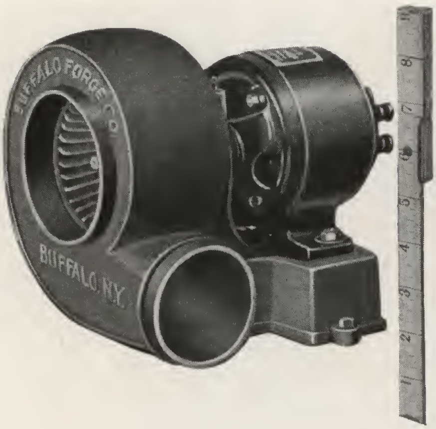
The operation of these fans is noiseless and due to their compactness can be installed in places where otherwise it would be impossible to secure ventilation. They have been used to great advantage in factories, boiler and engine rooms, lavatories, hotels, restaurants, theaters, etc. One of these outfits can easily be installed, so as to be completely concealed and not effect its operation or efficiency.

Size	Normal Speed	Cu. Ft. Air per Minute	Horse Power Required	Size of Pulley	Weight Lbs.
18	1000	2200	0.25	2 x 4	75
24	800	4000	0.50	2 x 4	100
30	650	6200	0.75	2¾x 6	170
36	525	8800	1.00	3 x 7	230
42	450	12000	1.50	3⅛x 8	325
48	400	18000	2.00	4 x 9	445
54	350	21000	2.50	4 x 9	560
60	320	25000	3.00	5 x 10	630
72	265	36000	5.00	5½x 12	820
84	227	50000	7.50	6 x 14	990

BABY CONOIDAL FANS

Ventilating and removing Smoke, Steam, Fumes and Gases in:

Banquet Rooms
Club Rooms
Dining Rooms
Kitchens
Restaurants
Laboratories
Lavatories
Workrooms
Telephone Booths
Staterooms
Cabins
Tunnels
Etc., Etc.



Blowing and Cooling:

Feathers
Fire Places
Gas Burners
Grates
Furnaces
Boiler Rooms
Candy Molds
Conduits
Engine Rooms
Motors

Evaporating and Drying in:

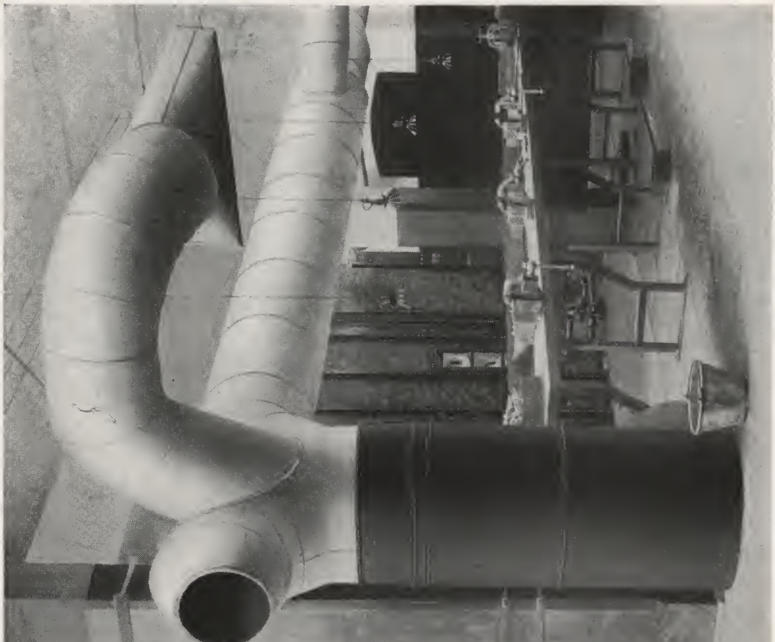
Barber Shops
Dry Cleaning Establishments
Hair Dressing Parlors
Photographic Studios
Etc., Etc.

The Baby Conoidal is of the high efficiency multiblade fan type. The wheel is so well balanced and the fan runs so smoothly and quietly at a relatively low speed, that practically noiseless operation is assured. These fans deliver a large volume of air at a low pressure and are unexcelled for general drying purposes, and for furnishing fresh, cool air to private homes, offices, telephone booths, staterooms, railroad cars, etc. These fans can also be used to exhaust hot air, smoke, fumes, etc., with equally good results.

A very important and convenient feature of this outfit is that the fan case may be swung around so as to discharge air in any direction or at any angle desired.

Cord and plug are furnished with each outfit, so that it can be attached to an electric light socket without any expense for installing.

Size	Capacity Air per Minute	Horse Power	Speed Rev. per Minute	Shipping Weight	Height
No. 1	90 cu. ft.	1/30	1800	25 lbs.	8¼ in.
No. 2	250 cu. ft.	⅛	1800	45 lbs.	10½ in.
No. 3	500 cu. ft.	¼	1800	65 lbs.	15 in.



PLANOIDAL STEEL PLATE FAN AND ENGINE INSTALLED FOR WARNER INSTRUMENT COMPANY, BELOIT, WIS.

PART FOUR

DATA ON HEATING AND VENTILATING

Relation of Velocity to Pressure

The laws governing the flow of air are perhaps less understood than almost any other branch of engineering data. The flow of air under high pressures must necessarily be investigated thermodynamically and the formula is therefore complicated. For ordinary fan work, however, where air is at low pressure, the comparison is so slight that it may be neglected with but little error and the same formula may be applied to the flow of air as to the flow of water, namely,

$$v \doteq \sqrt{2\,g\,h} = k\sqrt{\frac{p}{d}}$$

v = the velocity in feet per second,

Where p is the pressure in inches of water, and

d = the weight of air in pounds per cu. ft., which depends upon the temperature, humidity, and the barometric pressure.

d may be computed by the formula

$$d = k\,m\frac{H}{460 + T}$$

Where H is the barometric pressure in inches of mercury,

T is the temperature in degrees F., and

m is the factor to be obtained from a hygrometric table which shows the percentage of decrease in the density due to the presence of moisture as indicated by the depression of the wet bulb thermometer.

A formula has been given by some writers as

$$(1) \qquad v = k\sqrt{\frac{p}{(235 + p)\,d}}$$

This is supposed to make correction for the effect of compression which, however, it does not do, but gives a result which is even more in error than the approximate equation given above.

CORRESPONDING PRESSURES AND VELOCITIES OF DRY AIR AT 70° AND 29.92″ BAROMETER

Inches of Water	Ounces per Sq. In.	Velocity Ft. per Min.	Inches of Water	Ounces per Sq. In.	Velocity Ft. per Min.
.05	.0289	896	4.77	2.750	8745
.10	.0577	1266	5.00	2.884	8943
.20	.1154	1791	5.20	3.000	9134
.25	.1443	2003	5.50	3.172	9392
.30	.1730	2193	6.00	3.460	9810
.40	.2308	2533	6.07	3.500	9864
.43	.2500	2637	6.50	3.749	10210
.50	.2884	2832	6.94	4.000	10545
.60	.3460	3102	7.00	4.037	10595
.70	.4037	3351	7.50	4.326	10968
.75	.4326	3468	7.80	4.500	11187
.80	.4614	3582	8.00	4.614	11328
.87	.5000	3729	8.67	5.000	11792
.90	.5190	3800	9.00	5.190	12015
1.00	.5768	4005	9.54	5.500	12367
1.25	.7209	4478	10.00	5.768	12665
1.30	.7500	4566	10.40	6.000	12915
1.50	.8650	4905	11.00	6.344	13282
1.73	1.0000	5273	11.27	6.500	13445
1.75	1.0092	5298	12.00	6.921	13875
2.00	1.1535	5664	12.14	7.000	13950
2.17	1.2500	5895	13.00	7.497	14440
2.25	1.2975	6007	13.87	8.000	14913
2.50	1.4418	6332	14.00	8.074	14985
2.60	1.5000	6457	15.00	8.650	15510
2.75	1.5860	6641	15.61	9.000	15820
3.00	1.7300	6937	16.00	9.227	16020
3.03	1.7500	6976	17.00	9.805	16513
3.25	1.8740	7220	17.34	10.000	16675
3.47	2.0000	7457	18.00	10.380	16990
3.50	2.0185	7492	19.00	10.960	17456
3.75	2.1630	7756	19.07	11.000	17488
3.90	2.2500	7910	20.00	11.535	17910
4.00	2.3070	8010	20.81	12.000	18265
4.25	2.4510	8256	22.54	13.000	19012
4.34	2.5000	8337	24.28	14.000	19730
4.50	2.5950	8496	26.01	15.000	20420
4.75	2.7395	8729	27.74	16.000	21090

To obtain a more correct formula which will apply to higher pressures up to ½ of an atmosphere, we may assume the air is discharged under isothermal expansion, when we obtain the formula

$$(2) \qquad v_0 = k\sqrt{\frac{1}{d}}\sqrt{\log_{10}\frac{P_0 + P}{P_0}}$$

Where P_0 is the barometric pressure in pounds per sq. in.

P the pressure of the air above atmospheric pressure expressed in inches of mercury.

d the density in pounds per cu. ft.

If a slightly more exact expression is required, which allows for the adiabatic expansion, the thermodynamic equation is used which gives,

$$(3) \qquad v_0 = 109.2\sqrt{T_1 \left\{ 1 - \left(\frac{P_0}{P_0 + P}\right)^{.29} \right\}}$$

This latter formula is inconvenient in application, and varies so little from formula No. 2 with pressures under 1 pound per square inch that formula No. 2 is always preferable. Below is given a table showing the theoretical velocities corresponding to the various pressures.

CORRESPONDING VELOCITY FOR DRY AIR AT VARIOUS PRESSURES AND TEMPERATURES

Pressure		50°	60°	70°	100°	150°	300°	500°	550°
Inches	Ounces								
.25	.1443	1965	1986	2003	2059	2149	2399	2696	2895
.5	.2884	2778	2808	2832	2911	3038	3391	3812	4095
.75	.4326	3402	3439	3468	3565	3720	4153	4668	5020
1.0	.5768	3929	3971	4005	4117	4296	4796	5390	5795
1.25	.7209	4393	4440	4478	4602	4804	5362	6027	6470
1.50	.8650	4812	4864	4905	5042	5262	5874	6602	7100
1.75	1.0092	5197	5254	5298	5446	5683	6344	7131	7655
2.00	1.1535	5556	5616	5664	5822	6076	6783	7624	8195
2.25	1.2975	5892	5956	6007	6174	6443	7193	8085	8690

Very frequently in testing the capacity of air moving machinery under pressure, it becomes necessary to measure the delivery of the air through orifices of different forms. In such cases account must be taken of the effect of the *vena contracta* which is dependent upon the shape of orifice used. Various investigators have made experimental determinations of the coefficients of nozzles with air and it has been found at low pressures that the coefficient of discharge with air at low pressure varies but little from coefficient for flow of water.

Coefficient for sharp orifice in thin plate .60.

Coefficient for short length of straight pipe of uniform diameter, .825.

Measurement of Air Flow

The quantity and velocity of air discharged by a fan or flowing through a pipe may be determined by an anemometer, an orifice, a short length of pipe, a converging nozzle, or a pitot tube.

1. The anemometer is used in many cases where extreme accuracy is not required or where the velocity of the air is low, and is especially adaptable to measuring air entering a room through a register or duct. The space from which the air is flowing should be divided in several small squares, a velocity reading taken for each division, and the average result used. An anemometer should be frequently calibrated, as hot air will quickly dry up the bearings, and the readings will be effected as much as 10%. This instrument may also vary at high velocities.

2. Air may be measured by discharging into a large air-tight chamber, which can be easily provided for permanent testing work, from which the air escapes through cylindrical or conical orifices. The pressure in the chamber is recorded by a pressure gauge and the quantity of air delivered can be computed, using the coefficient of orifice given in the table below. This method gives static pressure and not total pressure.

COEFFICIENTS OF DISCHARGE

Angles of Convergence	Coefficient of Discharge	Angles of Convergence	Coefficient of Discharge
0° 0′	0.825	13° 24′	0.946
1 36	0.866	14 28	0.941
3 10	0.895	16 36	0.938
4 10	0.912	19 28	0.924
5 26	0.924	21 0	0.918
7 52	0.929	23 0	0.913
8 58	0.934	29 58	0.896
10 20	0.938	40 20	0.869
12 4	0.942	48 50	0.847

In all cases in using the above table, pressures should be taken in the chamber from which the air is discharged.

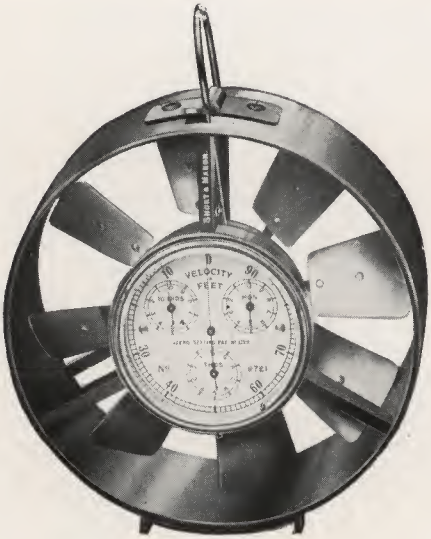
3. The fan may discharge into an air-tight chamber, the air escaping through a short length of pipe, and the static pressure be accurately measured. A coefficient of 0.825 should be applied to the area of the short pipe to determine the effective area and the velocity of the air.

4. In commercial work where a test is wanted before the fan is installed, a converging nozzle may be attached directly to the fan outlet, and the velocity pressure of air measured by a pitot tube. A proper coefficient of contraction, as shown by the table below, should be used, due to the convergence of the nozzle.

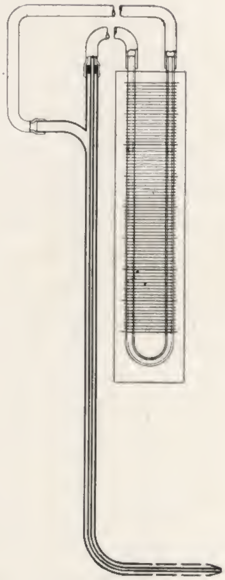
COEFFICIENTS OF CONTRACTION

Angles of Convergence	Coefficient of Contraction	Angles of Convergence	Coefficient of Contraction
7° 52′	0.998	23° 0′	0.937
8 58	0.992	29 58	0.919
10 20	0.987	40 20	0.887
12 4	0.986	48 50	0.861

5. A pitot tube is one of the most common instruments used to determine velocity and quantity of air flowing through a pipe. A typical form of the pitot tube is shown on page 94. This has two tubes, one inside the other. The outer tube has several small openings in the short bent portion which records static pressure. The inner bent tube has an opening in the end which records total pressure. By properly connecting the two tubes, the difference, or velocity pressure, may be read directly. In testing a fan the pitot tube should be placed 10 to 20 diameters from the fan outlet, and the air pipe should be the same diameter as the fan outlet. Velocity pressure in the center of the duct will be higher than the average, and the reading at the center of a round pipe should be multiplied by a coefficient of .80 to get true average velocity pressure. For more accurate work in round pipes and for square or rectangular pipes, a traverse of the pipe should be taken and an average of all the readings used. When the pipe is square or rectangular, it may be divided into a number of small squares or rectangles, a reading taken in the center of each, and the average result will be the true velocity pressure. If the pipe is round, it should be divided into a number of concentric rings of equal areas, and four readings taken in each area, horizontally and vertically across the pipe.

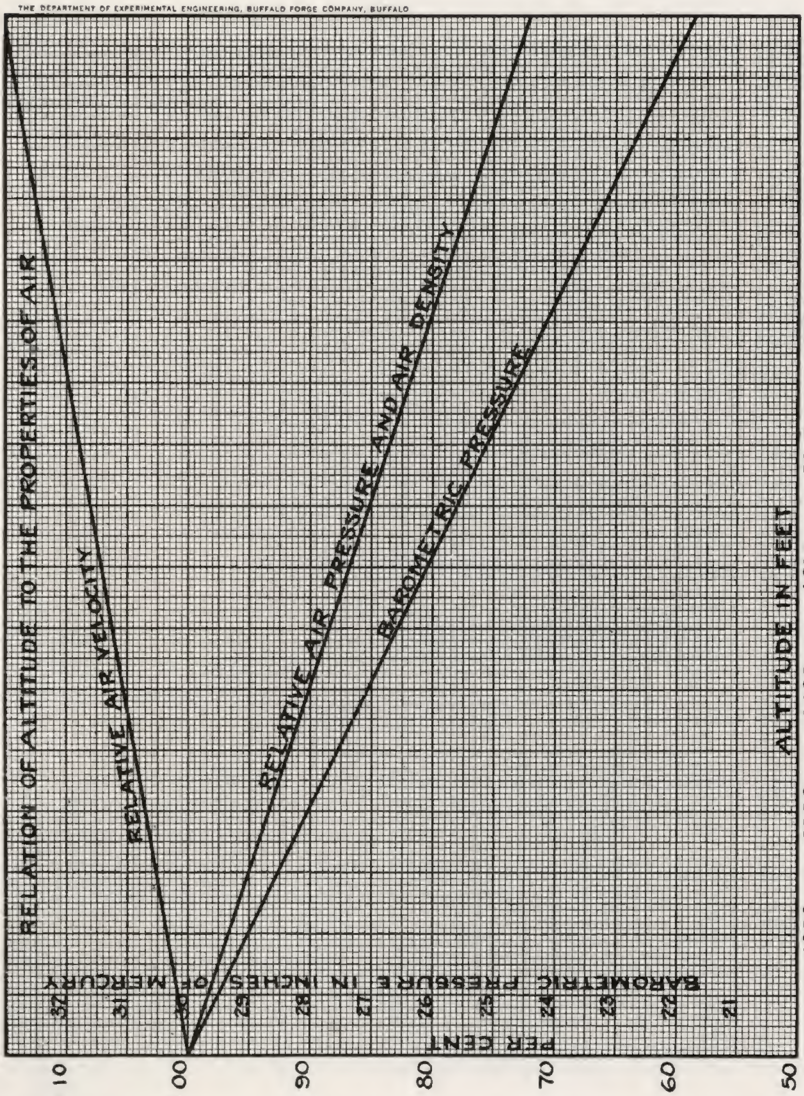


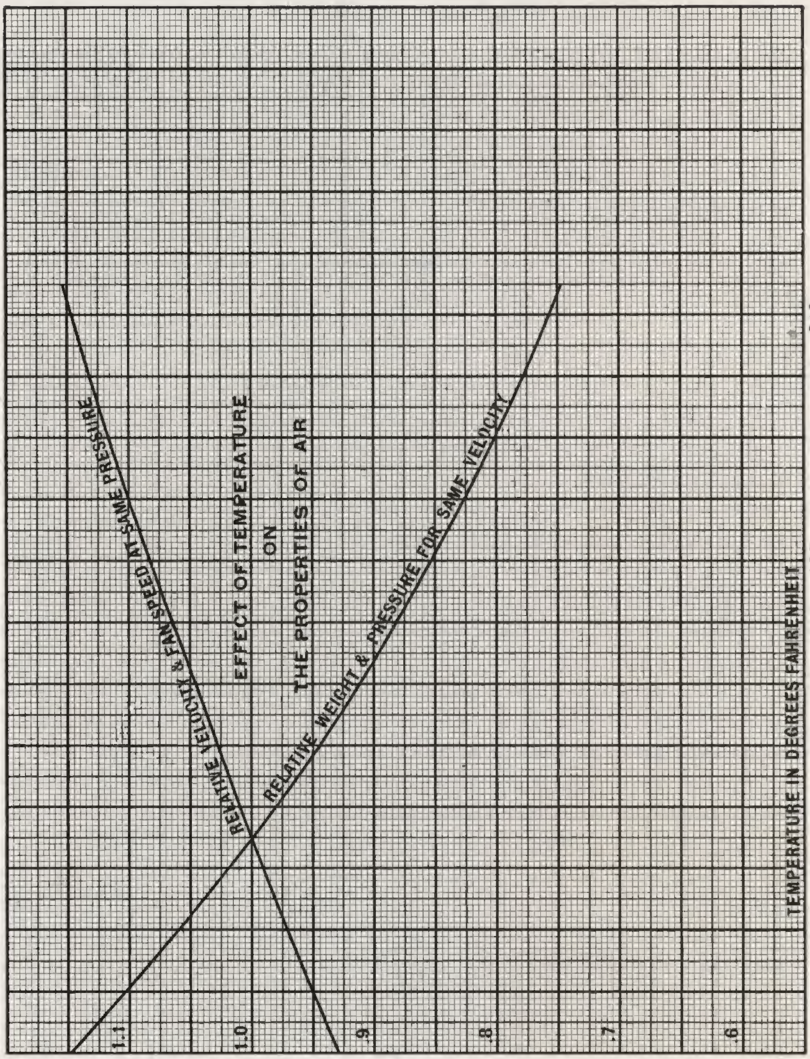
ANEMOMETER FOR MEASURING
VELOCITY OF AIR



PITOT TUBE

Much more complete data as to various methods of testing fans, together with methods of laying out a traverse, and other useful information and formulae, is given in the Engineering Hand Book, published by the Buffalo Forge Company, this Hand Book being a complete treatise on all branches of Fan Engineering.


THE DEPARTMENT OF EXPERIMENTAL ENGINEERING, BUFFALO FORGE COMPANY, BUFFALO
RELATION OF ALTITUDE TO THE PROPERTIES OF AIR
RELATIVE AIR VELOCITY
RELATIVE AIR PRESSURE AND AIR DENSITY
BAROMETRIC PRESSURE
ALTITUDE IN FEET
1000
2000
3000
4000
5000
6000
7000
8000
BAROMETRIC PRESSURE IN INCHES OF MERCURY
32
31
30
29
28
27
26
25
24
23
22
21
PER CENT
110
100
90
80
70
60
50


EFFECT OF TEMPERATURE
ON
THE PROPERTIES OF AIR
RELATIVE VELOCITY & FAN SPEED AT SAME PRESSURE
RELATIVE WEIGHT & PRESSURE FOR SAME VELOCITY
TEMPERATURE IN DEGREES FAHRENHEIT
0
20
40
60
80
100
120
140
160
180
200
220
240
1.1
1.0
.9
.8
.7
.6

Friction of Piping

A subject of great practical importance in fan work is the loss of pressure by friction in conveying air through piping. The expression for the flow of air in smooth circular metal pipes may be taken as approximately

$$F = \frac{l}{50\ d}\left(\frac{V}{(4005)}\right)^2$$

Where F is the loss of pressure in inches of water

V is the velocity in feet per minute

l is the length of the pipe in feet

d is the diameter of the pipe in feet, i. e., $\frac{l}{d}$=length of the pipe in diameters.

From this formula it will be seen that 50 diameters of smooth pipe produce a loss which corresponds to the velocity head. This formula is of the same general form developed by Weisbach but recent experiments have shown his coefficient to be considerably too high for smooth pipe and in this formula it has been corrected accordingly. For pipes with rough or uneven surfaces the coefficients must be increased accordingly. For tile and brick ducts we recommend that the coefficient be increased 25%. On pages 100 and 101 are given revised tables of piping friction which will be found very useful in estimating friction losses. For determining the carrying capacity of the pipe at any specified velocity flow, the tables on pages 98 and 99 may be used. To facilitate calculation of piping sizes for equal friction, the chart and table on pages 102 and 103 have been prepared. Table on page 103 shows the relative carrying capacity of round and rectangular pipe.

LEFT-HAND BOTTOM HORIZONTAL DISCHARGE PLANOIDAL FAN

CARRYING CAPACITY OF PIPES

This table specifies the diameters of pipes required for the passage of stated volumes of air at given velocities. The column, "Cubic feet of air per minute," indicates various quantities of air to be moved per minute. The figures at top of table give the velocities in feet per minute at which the air is to be moved, and the figures in the body of the table state the required diameters of pipes for the passage of the volumes mentioned at the given velocities.

DIAMETER OF PIPE IN INCHES

Cubic Feet of Air per Minute	Velocities											
	500	600	800	1000	1200	1500	1800	2000	2500	3000	3500	4000
200	9	8	7	7	6	6	6	6	6	6	6	6
400	13	11	10	9	8	8	7	7	6	6	6	6
600	15	14	12	11	10	9	8	8	7	7	6	6
800	18	16	14	13	12	10	9	9	8	8	7	7
1000	20	18	16	14	13	12	10	10	9	8	8	7
1200	21	20	17	15	14	13	11	11	10	9	9	8
1400	23	21	18	16	15	14	12	12	11	10	9	9
1600	25	23	20	18	16	15	13	13	11	11	10	9
1800	26	24	21	19	17	15	14	13	12	11	10	10
2000	28	25	22	20	18	16	15	14	13	12	11	10
2200	29	27	23	21	19	17	15	15	13	12	11	11
2400	30	28	24	21	20	18	16	15	14	13	12	11
2600	31	29	25	22	20	18	17	16	15	13	12	11
2800	33	30	26	23	21	19	18	16	15	14	13	12
3000	34	31	27	24	22	20	18	17	15	14	13	12
3200	34	32	28	25	23	20	19	18	15	15	13	13
3400	36	33	28	25	23	21	19	18	16	15	14	13
3600	37	34	29	26	24	21	20	19	16	15	14	13
3800	38	35	30	27	25	22	21	19	17	16	15	14
4000	39	35	31	28	25	22	21	20	18	16	15	14
4200	40	36	32	28	26	23	21	20	18	16	15	14
4400	41	37	32	29	26	24	22	21	18	17	16	15
4600	42	38	33	30	27	24	22	21	19	17	16	15
4800	42	39	34	30	28	25	22	21	19	18	16	15
5000	43	40	34	31	28	25	23	22	20	18	17	16
5200	44	40	35	31	29	25	24	22	20	18	17	16
5400			35	32	29	26	24	23	21	18	18	16
5600			36	33	30	27	24	23	21	19	18	17
5800			37	33	30	27	25	24	21	19	18	17
6000			38	34	31	28	25	24	21	20	18	17
6200			38	34	31	28	25	24	21	20	18	17
6400			39	35	32	28	26	25	22	20	19	18
6600			39	36	32	29	26	25	22	21	19	18
6800			40	36	33	29	27	25	23	21	19	18
7000			40	36	33	30	27	26	23	21	19	18
7200			41	37	34	30	28	26	23	21	20	19
7400			41	37	34	30	28	27	24	21	20	19
7600			42	38	34	31	28	27	24	22	20	19
7800			43	38	36	31	29	27	24	22	21	19
8000			43	39	36	32	29	28	25	22	21	20
8200				39	36	32	29	28	25	23	21	20
8400				40	36	33	30	28	25	23	21	20

CARRYING CAPACITY OF PIPES (CONTINUED)

DIAMETER OF PIPE IN INCHES

Cubic Ft. of Air per Minute	Velocities 1000	1200	1500	1800	2000	2500	3000	3500	4000
8600	40	37	33	30	29	25	23	21	20
8800	41	37	33	30	29	26	24	22	21
9000	41	38	34	31	29	26	24	22	21
9200	41	38	34	31	30	26	24	22	21
9400	42	38	34	31	30	27	24	22	21
9600	42	39	35	32	30	27	25	23	21
9800	43	39	36	32	30	27	25	23	21
10000	43	40	36	32	31	28	25	23	22
11000	45	41	37	33	31	29	26	24	23
12000	47	43	39	35	34	30	28	25	24
13000	49	45	40	37	35	31	29	27	25
14000	51	47	42	38	36	33	30	28	26
15000	53	48	43	40	38	34	31	28	27
16000	55	50	45	41	39	35	32	29	28
17000	56	51	46	42	40	36	33	30	28
18000	58	53	47	43	41	37	34	31	29
19000	60	54	49	44	42	38	34	32	30
20000	61	56	50	46	43	39	35	33	31
21000	63	57	51	47	44	40	36	34	31
22000	64	58	52	48	45	41	37	34	32
23000	65	60	53	49	46	42	38	35	33
24000	67	61	55	50	47	42	39	36	34
25000	68	62	56	51	48	43	40	37	34
26000	70	63	57	52	49	44	40	38	35
27000	71	65	58	53	50	45	41	38	36
28000	72	66	59	54	51	46	42	39	36
29000	73	67	60	55	52	47	42	39	37
30000	75	68	61	56	53	47	43	40	38
31000	76	69	62	57	54	48	44	41	38
32000	77	70	63	57	55	49	45	41	39
33000	78	72	64	58	56	50	45	42	39
34000	79	73	65	59	56	50	46	43	40
35000	81	74	66	60	57	51	47	43	40
36000	82	75	67	61	58	52	47	44	41
37000	83	76	68	62	59	52	48	44	42
38000	84	77	69	63	60	53	49	45	42
39000	85	78	70	63	60	54	49	46	43
40000	86	79	71	64	61	55	50	46	43
41000	87	79	71	65	62	55	50	47	44
42000	88	81	72	66	63	56	51	47	44
43000	89	82	73	66	63	57	51	48	44
44000	90	82	74	67	64	57	52	48	45
45000	91	83	75	68	65	58	53	49	46
46000	93	84	75	69	65	59	53	50	46
47000	93	85	76	70	66	59	54	50	47
48000	95	86	77	70	67	60	55	50	47
49000	95	87	78	71	68	60	55	51	48
50000	96	88	79	72	68	61	56	51	48
51000	97	89	79	73	69	62	56	52	49
52000	98	90	80	73	70	62	57	53	49
53000	99	90	81	74	70	63	57	53	50

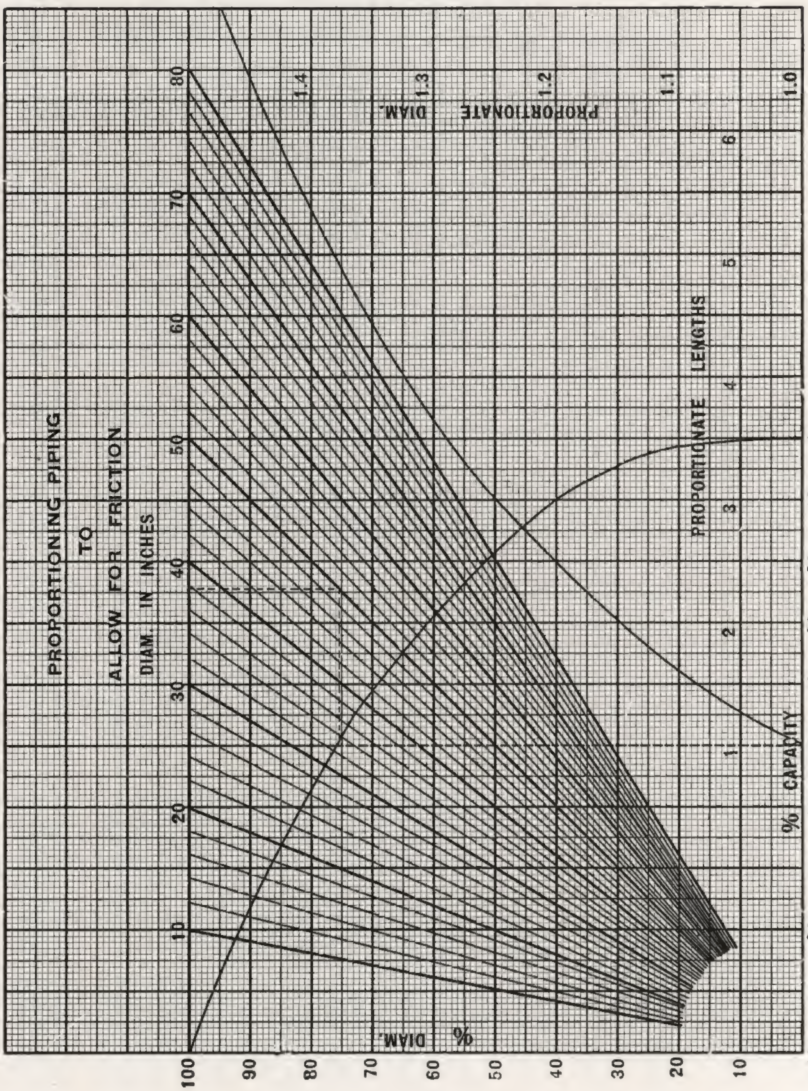
Cubic Ft. of Air per Minute	Velocities 1200	1500	1800	2200	2500	3000	3500	4000
54000	91	82	75	68	63	58	54	50
55000	92	82	75	68	64	58	54	51
56000	93	83	76	69	65	59	55	51
57000	94	84	77	69	65	60	55	52
58000	95	85	77	70	66	60	56	52
59000	95	85	78	71	66	60	56	52
60000	96	86	79	71	67	61	57	53
61000	97	87	79	72	67	62	57	53
62000	98	88	80	72	68	62	57	54
63000				73	68	63	58	54
64000				73	69	63	58	55
65000				74	70	63	59	55
66000				75	70	64	59	56
67000				75	71	64	60	56
68000				76	71	65	60	56
69000				76	71	65	61	57
70000				77	72	66	61	57
71000				77	73	66	61	57
72000				78	73	67	62	58
73000				78	74	67	62	58
74000				79	74	68	63	59
75000				79	75	68	63	59
76000				80	75	69	64	60
77000				81	76	69	64	60
78000				81	76	70	64	60
79000				82	77	70	65	61
80000				82	77	70	65	61
81000				83	78	71	66	61
82000				83	78	71	66	62
83000				84	79	72	66	62
84000				84	79	72	67	63
85000				85	79	73	67	63
86000				85	80	73	68	63
87000				86	80	73	68	64
88000				86	81	74	68	64
89000				87	81	74	69	64
90000				87	82	75	69	65
91000				88	82	75	70	65
92000				88	83	75	70	65
93000				88	83	76	70	66
94000				89	84	76	71	66
95000				89	84	77	71	66
96000				90	84	77	71	67
97000				90	85	77	72	67
98000				91	85	78	72	68
99000				91	86	78	72	68
100000				92	86	79	73	68

TABLE EXHIBITING PRESSURE REQUIRED TO OVERCOME FRICTION OF AIR PASSING THROUGH PIPES

Velocity of Air in Ft. per Min.	Loss of Pressure per 100 Ft. in Inches of Water — Diameter of Pipe in Inches													
	3 in.	4 in.	5 in.	6 in.	7 in.	8 in.	9 in.	10 in.	12 in.	14 in.	16 in.	18 in.	20 in.	22 in.
200	.026	.019	.016	.012	.010	.009	.008	.007	.007	.005	.005	.003	.003	.003
300	.057	.043	.035	.029	.024	.023	.019	.017	.014	.012	.010	.010	.009	.009
400	.102	.076	.062	.050	.043	.038	.033	.031	.026	.022	.019	.017	.016	.014
500	.161	.120	.097	.080	.069	.061	.054	.049	.040	.035	.029	.027	.024	.022
600	.231	.173	.139	.116	.099	.087	.076	.069	.057	.050	.043	.038	.035	.031
700	.314	.239	.189	.158	.135	.118	.104	.094	.078	.068	.059	.052	.047	.043
800	.411	.309	.246	.206	.177	.154	.137	.123	.102	.088	.076	.069	.062	.056
900	.520	.390	.312	.260	.224	.194	.173	.156	.130	.111	.097	.087	.078	.071
1000	.642	.482	.385	.321	.276	.241	.213	.192	.160	.137	.120	.108	.097	.088
1500	1.444	1.083	.867	.723	.619	.541	.482	.434	.361	.312	.277	.243	.225	.198
2000	2.568	1.927	1.542	1.285	1.101	.964	.855	.770	.642	.550	.482	.428	.385	.350
2500	4.013	3.004	2.409	2.006	1.748	1.505	1.337	1.205	1.004	.860	.753	.669	.603	.548
3000	5.774	4.335	3.468	2.890	2.478	2.168	1.927	1.734	1.444	1.238	1.084	.964	.867	.789
3500	7.872	5.902	4.722	3.820	3.373	2.956	2.624	2.360	1.966	1.685	1.476	1.311	1.179	1.073
4000	10.276	7.706	6.166	5.138	4.405	3.853	3.425	3.083	2.568	2.202	1.926	1.713	1.542	1.401
4500	13.005	9.754	7.803	6.560	5.573	4.878	4.335	3.728	3.251	2.787	2.438	2.168	1.951	1.774
5000	16.055	12.051	9.634	8.084	6.880	5.934	5.351	4.852	4.014	3.440	3.010	2.676	2.409	2.190
5500	20.643	14.577	11.656	9.713	8.340	7.288	6.477	5.827	4.857	4.162	3.642	3.237	2.913	2.648
6000	23.120	17.340	13.871	11.561	9.908	8.670	7.706	6.936	5.780	4.985	4.335	3.853	3.468	3.152

TABLE EXHIBITING PRESSURE REQUIRED TO OVERCOME FRICTION OF AIR PASSING THROUGH PIPES (CONTINUED)

Velocity of Air in Ft. per Min.	Loss of Pressure per 100 Ft. in Inches of Water — Diameter of Pipe in Inches											
	24 in.	26 in.	28 in.	30 in.	34 in.	38 in.	42 in.	46 in.	50 in.	54 in.	58 in.	62 in.
200	.00322	.00296	.00274	.00257	.00225	.00205	.00184	.00166	.00156	.00139	.00139	.00121
300	.00711	.00668	.00619	.00577	.00510	.00456	.00413	.00376	.00347	.00329	.00295	.00277
400	.01281	.01183	.01099	.01025	.00905	.00810	.00732	.00668	.00607	.00572	.00538	.00486
500	.02005	.01850	.01719	.01604	.01415	.01266	.01146	.01046	.00954	.00884	.00815	.00763
600	.02890	.02667	.02476	.02311	.02039	.01826	.01651	.01491	.01387	.01283	.01179	.01127
700	.03929	.03628	.03388	.03144	.02773	.02481	.02245	.02046	.01873	.01751	.01630	.01526
800	.05134	.04741	.04401	.04108	.03624	.03243	.02934	.02670	.02462	.02289	.02133	.01994
900	.06503	.06003	.05571	.05202	.04590	.04106	.03716	.03399	.03121	.02878	.02688	.02514
1000	.08021	.07404	.06876	.06417	.05661	.05067	.04583	.04214	.03850	.03555	.03312	.03104
1500	.18064	.16677	.15482	.14450	.12750	.11409	.10320	.09427	.08653	.08010	.07473	.06988
2000	.32105	.29638	.27271	.25451	.22460	.20092	.18182	.16732	.15417	.14270	.13282	.12415
2500	.50129	.46300	.42995	.40129	.35402	.31678	.28660	.26167	.24069	.22281	.20740	.19403
3000	.72250	.66695	.61930	.57800	.51000	.45631	.41270	.37680	.34681	.32096	.29895	.27970
3500	.98330	.90761	.84282	.78661	.69415	.62102	.56190	.51295	.47181	.43700	.40680	.38051
4000	1.2841	1.1853	1.1006	1.0274	.90650	.81111	.73381	.66985	.61575	.57066	.53131	.49696
4500	1.6257	1.5051	1.3934	1.3050	1.1476	1.0267	.92899	.84809	.78032	.72135	.67106	.62925
5000	2.0068	1.8525	1.7201	1.5986	1.4166	1.2309	1.1467	1.0462	.96337	.89178	.83022	.77666
5500	2.4284	2.2411	2.0814	1.9426	1.7140	1.5318	1.3873	1.2667	1.1654	1.0791	1.0046	.93980
6000	2.8900	2.6611	2.4771	2.3121	2.0402	1.8252	1.6473	1.5078	1.3872	1.2844	1.1947	1.1167


PROPORTIONING PIPING
TO
ALLOW FOR FRICTION
DIAM. IN INCHES
PROPORTIONATE DIAM.
PROPORTIONATE LENGTHS
% CAPACITY
% DIAM.

CIRCULAR EQUIVALENTS OF RECTANGULAR DUCTS

Side Rect. Duct.	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25
3	3.3																						
4	3.8	4.4																					
5	4.3	4.9	5.5																				
6	4.7	5.4	6.0	6.6																			
7	5.0	5.8	6.5	7.0	7.7																		
8	5.2	6.1	6.9	7.6	8.2	8.8																	
9	5.5	6.5	7.3	8.0	8.7	9.3	9.9																
10	5.8	6.8	7.7	8.4	9.2	9.8	10.4	11.0															
11	6.1	7.1	8.0	8.8	9.6	10.2	10.9	11.5	12.1														
12	6.4	7.4	8.3	9.2	10.0	10.7	11.4	12.0	12.6	13.2													
13	6.6	7.6	8.7	9.6	10.4	11.1	11.8	12.5	13.1	13.7	14.3												
14	6.8	7.9	8.9	9.9	10.8	11.5	12.3	12.9	13.6	14.3	14.9	15.4											
15	6.9	8.2	9.2	10.2	11.1	11.9	12.7	13.4	14.1	14.7	15.3	16.0	16.5										
16	7.1	8.4	9.5	10.5	11.4	12.3	13.1	13.8	14.5	15.2	15.8	16.5	17.1	17.6									
17	7.3	8.6	9.8	10.8	11.8	12.6	13.5	14.2	15.0	15.7	16.3	17.0	17.6	18.2	18.7								
18	7.5	8.9	10.0	11.1	12.1	13.0	13.8	14.6	15.4	16.1	16.8	17.4	18.1	18.7	19.2	19.8							
19	7.7	9.1	10.3	11.4	12.4	13.3	14.2	15.0	15.8	16.5	17.2	17.9	18.6	19.2	19.8	20.4	20.9						
20	7.9	9.3	10.5	11.6	12.7	13.6	14.5	15.4	16.2	17.0	17.6	18.4	19.0	19.7	20.3	20.9	21.5	22.0					
22	8.2	9.7	11.0	12.1	13.2	14.2	15.2	16.1	16.9	17.8	18.5	19.2	19.9	20.6	21.3	21.9	22.5	23.1	23.6	24.2			
24	8.5	10.0	11.4	12.6	13.8	14.8	15.8	16.8	17.6	18.5	19.3	20.0	20.8	21.5	22.2	22.8	23.5	24.0	24.7	25.2	25.9	26.4	
26	8.8	10.4	11.8	13.1	14.3	15.4	16.4	17.3	18.3	19.2	20.0	20.8	21.6	22.3	23.0	23.8	24.4	25.1	25.7	26.3	26.9	27.5	28.1
28	9.1	10.8	12.2	13.5	14.8	15.9	17.0	18.0	19.0	19.8	20.7	21.5	22.4	23.1	23.9	24.6	25.3	26.0	26.6	27.3	27.9	28.5	29.1
30	9.3	11.0	12.6	13.9	15.2	16.4	17.5	18.5	19.5	20.5	21.4	22.2	23.1	23.9	24.7	25.4	26.2	26.8	27.5	28.2	28.8	29.5	30.1
32	9.6	11.3	12.9	14.3	15.6	16.9	18.0	19.1	20.1	21.1	22.0	22.9	23.8	24.6	25.4	26.2	27.0	27.7	28.4	29.1	29.8	30.5	31.1
34	9.8	11.6	13.2	14.7	16.1	17.3	18.5	19.6	20.7	21.6	22.6	23.5	24.4	25.3	26.2	26.9	27.7	28.5	29.2	30.0	30.7	31.3	32.0
36	10.1	11.9	13.6	15.1	16.4	17.7	19.0	20.1	21.2	22.2	23.2	24.2	25.1	26.0	26.8	27.7	28.5	29.3	30.0	30.8	31.5	32.2	32.9
38	10.3	12.2	13.9	15.4	16.8	18.2	19.4	20.6	21.7	22.8	23.8	24.8	25.8	26.7	27.5	28.4	29.2	30.0	30.8	31.5	32.3	33.1	33.8
40	10.5	12.5	14.3	15.7	17.2	18.6	19.8	21.1	22.2	23.3	24.4	25.4	26.4	27.3	28.2	29.1	29.9	30.8	31.6	32.4	33.1	33.9	34.6
42	10.8	12.7	14.5	16.1	17.6	19.0	20.3	21.6	22.7	23.8	24.9	25.9	26.9	27.9	28.8	29.8	30.7	31.4	32.2	33.0	33.8	34.5	35.2
44	11.0	13.0	14.8	16.4	18.0	19.4	20.7	22.0	23.1	24.3	25.4	26.5	27.5	28.5	29.5	30.3	31.2	32.1	32.9	33.7	34.5	35.3	36.1
46	11.2	13.3	15.1	16.7	18.4	19.8	21.1	22.4	23.6	24.8	25.9	27.0	28.1	29.1	30.1	31.0	31.9	32.8	33.8	34.6	35.4	36.2	37.0
48	11.4	13.5	15.4	17.0	18.7	20.1	21.5	22.8	24.1	25.2	26.4	27.5	28.6	29.6	30.5	31.6	32.5	33.4	34.3	35.2	36.1	37.0	37.8
50	11.6	13.7	15.7	17.3	19.0	20.4	21.9	23.2	24.5	25.7	26.9	28.0	29.2	30.3	31.3	32.2	33.1	34.1	35.0	35.9	36.8	37.6	38.4
52	11.8	13.9	15.9	17.6	19.2	20.8	22.2	23.6	24.9	26.2	27.4	28.5	29.6	30.7	31.8	32.9	33.8	34.7	35.6	36.5	37.4	38.3	39.1
54	12.0	14.1	16.1	17.9	19.6	21.1	22.6	24.0	25.3	26.6	27.8	29.0	30.1	31.2	32.3	33.4	34.4	35.3	36.3	37.2	38.0	38.9	39.8
56	12.1	14.3	16.3	18.2	19.9	21.5	22.9	24.4	25.7	27.0	28.3	29.5	30.6	31.7	32.8	33.9	34.9	35.9	36.9	37.8	38.7	39.6	40.4
58	12.3	14.6	16.6	18.4	20.2	21.8	23.3	24.7	26.1	27.4	28.7	30.0	31.1	32.2	33.3	34.4	35.4	36.4	37.4	38.4	39.4	40.3	41.2
60	12.4	14.7	16.8	18.7	20.4	22.1	23.6	25.1	26.5	27.8	29.1	30.5	31.6	32.7	33.8	34.9	36.1	37.1	38.1	39.1	40.0	40.9	41.8
62	12.6	15.0	17.0	19.0	20.7	22.4	24.0	25.5	26.9	28.2	29.5	30.9	32.1	33.2	34.3	35.4	36.6	37.7	38.7	39.6	40.6	41.6	42.5
64	12.8	15.1	17.3	19.2	21.0	22.7	24.3	25.9	27.3	28.6	29.9	31.3	32.6	33.7	34.8	35.9	37.1	38.2	39.2	40.2	41.2	42.2	43.1
66	12.9	15.3	17.5	19.5	21.2	23.0	24.6	26.2	27.7	29.0	30.3	31.7	33.0	34.2	35.3	36.4	37.6	38.7	39.8	40.8	41.8	42.8	43.7
68	13.1	15.5	17.7	19.7	21.5	23.3	24.9	26.5	28.1	29.4	30.7	32.1	33.4	34.7	35.8	36.9	38.1	39.2	40.3	41.4	42.4	43.4	44.4

WEIGHT OF BLACK STEEL PIPES IN POUNDS (AVOR.) PER RUNNING FOOT

Dia. Pipe	Material Sq. Ft. per Running Ft.	NUMBER OF GAUGE, U. S. S.							Dia. Pipe	Material Sq. Ft. per Running Ft.	NUMBER OF GAUGE, U. S. S.						
		No. 24	No. 22	No. 20	No. 18	No. 16	No. 14	No. 12			No. 24	No. 22	No. 20	No. 18	No. 16	No. 14	No. 12
4	1.13	1.30	1.58	1.86	2.43	2.99	3.62	5.08	34	9.15	10.53	12.81	15.10	19.68	24.43	29.30	41.20
5	1.39	1.60	1.95	2.29	2.99	3.68	4.45	6.25	35	9.41	10.82	13.18	15.51	20.20	24.90	30.10	42.30
6	1.65	1.90	2.31	2.72	3.54	4.36	5.28	7.42	36	9.67	11.11	13.54	15.95	20.78	25.60	30.90	43.50
7	1.91	2.20	2.67	3.15	4.10	5.05	6.11	8.58	37	9.93	11.42	13.90	16.40	21.38	26.30	31.80	44.70
8	2.18	2.50	3.05	3.60	4.68	5.77	6.97	9.80	38	10.19	11.71	14.28	16.80	21.90	27.00	32.60	45.80
9	2.44	2.80	3.42	4.03	5.25	6.47	7.80	10.98	39	10.46	12.03	14.65	17.27	22.50	27.74	33.50	47.10
10	2.70	3.10	3.78	4.45	5.80	7.15	8.64	12.15	40	10.72	12.33	15.00	17.70	23.01	28.40	34.30	48.25
11	2.96	3.40	4.15	4.88	6.36	7.85	9.47	13.31	41	10.98	12.62	15.38	18.11	23.60	29.10	35.10	49.40
12	3.22	3.70	4.50	5.31	6.91	8.52	10.30	14.48	42	11.24	12.93	15.75	18.55	24.20	29.80	36.00	50.60
13	3.48	4.00	4.88	5.74	7.48	9.21	11.15	15.66	43	11.59	13.32	16.21	19.10	24.90	30.70	37.05	52.10
14	3.74	4.30	5.23	6.17	8.03	9.90	11.97	16.84	44	11.85	13.64	16.60	19.55	25.50	31.40	37.90	53.30
15	4.01	4.61	5.61	6.61	8.61	10.61	12.83	18.03	45	12.11	13.93	16.97	20.00	26.00	32.10	38.75	54.50
16	4.27	4.91	5.97	7.04	9.16	11.29	13.65	19.17	46	12.37	14.23	17.31	20.40	26.60	32.80	39.60	55.70
17	4.53	5.21	6.35	7.48	9.74	12.00	14.49	20.40	47	12.63	14.52	17.70	20.85	27.20	33.45	40.40	56.80
18	4.87	5.60	6.81	8.03	10.45	12.89	15.55	21.90	48	12.90	14.83	18.07	21.30	27.75	34.20	41.30	58.00
19	5.14	5.91	7.20	8.48	11.04	13.60	16.42	23.10	49	13.15	15.11	18.40	21.70	28.25	34.80	42.10	59.20
20	5.40	6.21	7.56	8.90	11.60	14.30	17.26	24.30	50	13.41	15.42	18.80	22.15	28.80	35.55	42.90	60.40
21	5.59	6.43	7.83	9.22	12.00	14.80	17.87	25.10	51	13.66	15.71	19.13	22.55	29.40	36.20	43.75	61.50
22	5.92	6.80	8.28	9.75	12.70	15.65	18.90	26.60	52	13.94	16.01	19.50	23.00	30.00	36.90	44.60	62.65
23	6.18	7.11	8.66	10.20	13.29	16.38	19.80	27.80	54	14.46	16.62	20.25	23.85	31.10	38.30	46.30	65.00
24	6.45	7.41	9.04	10.63	13.85	17.08	20.65	29.00	56	15.07	17.32	21.10	24.85	32.40	39.90	48.20	67.80
25	6.71	7.71	9.40	11.06	14.40	17.75	21.50	30.20	58	15.58	17.91	21.80	25.70	33.50	41.30	49.80	70.20
26	6.97	8.01	9.75	11.48	14.96	18.41	22.30	31.30	60	16.12	18.53	22.60	26.65	34.70	42.75	51.60	72.60
27	7.23	8.31	10.11	11.93	15.51	19.12	23.10	32.50	62	16.65	19.16	23.30	27.50	35.80	44.10	53.30	75.00
28	7.50	8.62	10.50	12.38	16.10	19.87	24.00	33.75	64	17.16	19.72	24.00	28.30	36.90	45.50	54.90	77.20
29	7.75	8.91	10.85	12.78	16.67	20.50	24.80	34.90	66	17.66	20.30	24.70	29.15	38.00	46.80	56.50	79.40
30	8.10	9.32	11.34	13.37	17.40	21.45	25.90	36.40	68	18.21	20.95	25.50	30.00	39.15	48.25	58.30	81.80
31	8.36	9.61	11.70	13.80	18.00	22.15	26.75	37.60	70	18.75	21.55	26.25	30.90	40.30	49.70	60.00	84.30
32	8.62	9.92	12.07	14.25	18.52	22.83	27.60	38.80	72	19.25	22.15	27.00	31.80	41.40	51.00	61.60	86.60
33	8.88	10.21	12.45	14.66	19.10	23.50	28.40	40.00	74	19.79	22.75	27.70	32.65	42.60	52.40	63.30	89.00

WEIGHT PER LINEAL FOOT FOR GALVANIZED IRON PIPES

U. S. Standard Gauge

Dia. of Pipe	Sq. Ft. per Running Ft.	Number of Gauge 26	24	22	20	18	16
4	1.13	1.13	1.47	1.69	1.97	2.56	3.10
5	1.39	1.39	1.80	2.08	2.43	3.19	3.82
6	1.65	1.65	2.14	2.47	2.89	3.79	4.54
7	1.91	1.91	2.48	2.86	3.34	4.39	5.25
8	2.18	2.18	2.83	3.27	3.81	5.01	6.00
9	2.44	2.44	3.17	3.66	4.27	5.61	6.71
10	2.70	2.70	3.51	4.05	4.72	6.21	7.42
11	2.96	2.96	3.85	4.44	5.18	6.80	8.14
12	3.22	3.22	4.18	4.83	5.63	7.40	8.85
13	3.48	3.48	4.52	5.22	6.09	8.00	9.57
14	3.74	3.74	4.86	5.61	6.54	8.60	10.28
15	4.01	4.01	5.21	6.01	7.01	9.22	10.86
16	4.27	4.27	5.55	6.40	7.47	9.82	11.74
17	4.53	4.53	5.85	6.79	7.92	10.42	12.45
18	4.87	4.87	6.33	7.30	8.51	11.18	13.36
19	5.14	5.14	6.68	7.71	9.00	11.80	14.11
20	5.40	5.40	7.02	8.10	9.45	12.42	14.85
21	5.59	5.59	7.26	8.39	9.78	12.85	15.36
22	5.92	5.92	7.70	8.88	10.35	13.60	16.25
23	6.18	6.18	8.04	9.27	10.81	14.40	17.00
24	6.45	6.45	8.38	9.67	11.30	14.84	17.71
25	6.71	6.71	8.72	10.06	11.74	15.41	18.41
26	6.97	6.97	9.05	10.45	12.20	16.00	19.15
27	7.33	7.33	9.40	10.85	12.67	16.62	19.87
28	7.50	7.50	9.75	11.27	13.13	17.26	20.60
29	7.75	7.75	10.07	11.63	13.58	17.81	21.30
30	8.10	8.10	10.54	12.17	14.20	18.62	22.25
31	8.36	8.36	10.87	12.54	14.63	19.20	23.00
32	8.62	8.62	11.20	12.93	15.10	19.84	23.70
33	8.88	8.88	11.56	13.34	15.56	20.42	24.40
34	9.15	9.15	11.90	13.73	16.00	21.08	25.18
35	9.41	9.41	12.23	14.10	16.48	21.65	25.85
36	9.67	9.67	12.57	14.50	16.91	22.22	26.60
37	9.93	9.93	12.91	14.90	17.40	22.84	27.30
38	10.19	10.19	13.25	15.29	17.81	23.40	28.00
Weights in Lbs. (Avor.) Per Running Ft.							

Dia. of Pipe	Sq. Ft. per Running Ft.	Number of Gauge 26	24	22	20	18	16
39	10.46	10.46	13.60	15.60	18.31	24.02	28.70
40	10.72	10.72	13.95	16.08	18.76	24.68	29.50
41	10.98	10.98	14.27	16.47	19.20	25.25	30.20
42	11.24	11.24	14.60	16.86	19.61	25.86	30.90
43	11.59	11.59	15.06	17.38	20.30	26.60	31.80
44	11.85	11.85	15.40	17.78	20.74	27.25	32.60
45	12.11	12.11	15.75	18.17	21.20	27.90	33.30
46	12.37	12.37	16.10	18.55	21.62	28.43	34.00
47	12.63	12.63	16.40	18.95	22.10	29.00	34.70
48	12.90	12.90	16.78	19.35	22.60	29.70	35.50
49	13.15	13.15	17.10	19.72	23.00	30.25	36.20
50	13.41	13.41	17.45	20.12	23.50	30.90	36.90
51	13.66	13.66	17.75	20.49	23.90	31.40	37.50
52	13.94	13.94	18.12	20.97	24.40	32.00	38.30
53	14.20	14.20	18.46	21.30	24.90	32.66	39.00
54	14.46	14.46	18.80	21.69	25.30	33.20	39.70
55	14.81	14.81	19.28	22.22	25.94	34.10	40.80
56	15.07	15.07	19.60	22.61	26.40	34.65	41.40
57	15.33	15.33	19.95	23.00	26.80	35.21	42.10
58	15.58	15.58	20.30	23.37	27.30	35.84	42.80
59	15.83	15.83	20.55	23.74	27.70	36.40	43.50
60	16.12	16.12	20.95	24.18	28.20	37.00	44.30
62	16.65	16.65	21.65	24.97	29.10	38.20	45.70
64	17.16	17.16	22.30	25.74	30.00	39.50	47.20
66	17.66	17.66	22.97	26.49	30.90	40.60	48.50
68	18.21	18.21	23.65	27.31	31.83	41.80	50.00
70	18.75	18.75	24.40	28.12	32.80	43.10	51.50
72	19.25	19.25	25.02	29.92	33.70	44.30	53.00
74	19.79	19.79	25.70	29.68	34.65	45.50	54.50
76					35.62	45.77	54.73
78					35.75	46.96	55.13
80					36.65	48.16	56.63
82					37.57	49.40	58.00
84					38.50	50.60	59.40
86					39.39	51.77	60.77
Weight Per Sq. Ft.		1.00	1.30	1.50	1.75	2.30	2.70

Heating and Ventilating
Ducts to 19″ Dia. 26 Ga.
" 20″ to 29″ " 24 "
" 30″ to 39″ " 22 "
" 40″ to 49″ " 20 "
" 50″ to 84″ " 18 "
Above 84″ " 16 "

For Planing-Mill Work
Ducts to 8″ Dia. 24 Ga.
" 9″ to 14″ " 22 "
" 15″ to 20″ " 20 "
" 21″ to 30″ " 18 "

DATA FOR DETERMINING SIZES OF MAIN AND BRANCH PIPES AND FLUES

Systems Most published rules involve arbitrary constants and tables without giving the basic formula or reasons in determining flue, register and pipe sizes. The most efficient arrangements can be made only when the hypothesis of calculation is understood. The essential data is here given and while its application requires more than merely taking sizes from tables, the whys and wherefores are known, and in this knowledge there is considerable satisfaction.

The piping systems for industrial buildings and those for public buildings are figured according to two distinct methods. In industrial buildings the problem is chiefly to convey the heat units with as great an economy of power, material and space as possible, while in public buildings there are the additional requirements of freedom from noise and prevention of drafts. In industrial buildings the air is usually conveyed through one or two main lines extending lengthwise of the building, the areas of such pipes decreasing as they extend, to give a uniform distribution of air throughout. On the other hand in public buildings, individual ducts are carried from the apparatus to each room, so that it is evident the same method is not applicable to both systems.

HOW TO PROPORTION PIPING IN INDUSTRIAL BUILDINGS

Method In proportioning the main and branch pipes in industrial buildings, the primary aim is to secure as uniform a distribution as possible without the necessity of dampering; secondly, to secure economy of power and economy of material. It has been found good practice in proportioning piping systems to decrease the velocity in the main pipes as the air quantity decreases. This principle of proportioning has three advantages. First: It utilizes the velocity of the air in producing static pressure in the system. Second: By this means a nearly uniform static pressure may be secured in all parts of the pipe line, giving a very uniform distribution of air throughout. Third: It reduces the friction in the smaller pipes, which would otherwise be excessive.

In carrying out this idea in the proportioning of the piping this company employs an original and accurate method. This method has been carefully tested and has been found to give an almost ideal

distribution; the principle involved is to so proportion the velocities in the various pipe sizes as to give equal friction in all air pipes per running foot regardless of their size. It may easily be shown that the equation which relates the carrying capacity of pipe to its size to suit this condition is:

$$\frac{d_2}{d_1} = \left(\frac{C_2}{C_1}\right)^{\frac{2}{5}}$$

Where d_1 and d_2 are the relative diameters of two pipes and C_1 and C_2 are the relative carrying capacities. As an equation in this form would be difficult of computation, the chart shown on page 102 is conveniently employed. In using this chart we start with the main pipe equal in area to the fan outlet, or 10 to 20% larger as circumstances may require. All sizes are proportioned directly from this main pipe size. It will be noted that the curve is plotted for per cent. capacity and for per cent. diameter according to the formula for constant friction per foot of length. For instance if we have a branch pipe which is required to carry 50% of the capacity of the main pipe, we find the point on the curve which corresponds to 50% capacity and which gives a corresponding point of 76% diameter; that is, a pipe to carry 50% of the capacity with the same friction per foot must have 76% of the diameter, which may be easily calculated or be read directly from the chart for various pipe sizes. It will be seen that straight lines are drawn for pipe sizes from 20″ up to 80″ in diameter. Supposing the size of the main pipe is 60″ in diameter, then following from left to right along the line of 76% diameter to the line of 60″ pipe we find from the scale above a diameter of 46″, which is the size of pipe which has half the capacity of 60″ pipe with the same friction per foot. By this method the sizes may be read off rapidly without any intermediate figuring whatever.

Application

Take the following example which shows the method: Let the main pipe from the fan be 48″ in diameter and suppose a straight duct having 10 equal outlets. The first section of piping is 48″, the second section has a capacity of 90%, the third section 80%, the fourth 70%, and so on; corresponding to 90% we find a diameter of 96% which for a 48″ pipe gives us 46″ for the second section. For the third section we have 80% capacity corresponding to 91% diameter, or again following from left to right to the 48″ line, we find a diameter of approximately 44″. For the fourth section we have 70% capacity with a corresponding pipe size of 86½% of the main pipe and a diameter of between 41″ and 42″

determined as before. For the last section we have 10% capacity or 40% diameter which gives a diameter of between 19″ and 20″. The outlets may of course be proportioned independently; the same is true of exceptionally long branches which after having been figured in the ordinary way should be increased by a certain percentage throughout as judgment may determine, to decrease the friction.

Determination of Friction

For perfectly smooth, straight galvanized iron pipe it has been found as stated above that the loss of pressure in a length equivalent to 50 diameters is approximately equal to the pressure corresponding to the velocity, i. e., to the velocity head. This holds true for all gases under usual velocities and also for water. In brick and concrete ducts, however, it is advisable to figure 25% more friction or in other words a loss in pressure corresponding to the velocity head for every 40 diameters, i. e., in a 12″ brick duct 40 feet long or 24″ brick duct 80 feet long, the loss in pressure will correspond to the velocity. For instance, 2000 velocity under those conditions will cause a loss in pressure of ¼ in. In addition to the above it is necessary to figure the loss in elbows. The factor for elbows is difficult to determine exactly, but from the best information obtainable it appears that one elbow with usual radius is equivalent to a length of pipe of approximately 10 diameters.

Now by the foregoing method of proportioning piping, it becomes unnecessary to figure the resistance of each section of pipe independently as the friction is constant per foot of length. It is simply necessary to know the length of the longest run of piping in feet, the number and sizes of elbows and the diameter and velocity in the largest pipe, as the loss is exactly the same as though the entire amount of air was carried through the largest pipe the entire distance. It is usual to figure the area of the largest pipe approximately equal to the area of the fan outlet. It should be noted that the velocity at the outlet of a Buffalo fan at the rated capacity is equal to one-half of the peripheral velocity, so that the velocity head in the main pipe will be $(½)^2=¼$ the total fan pressure. For convenience we may assume the fan to operate at 1 in., then the loss in piping thus proportioned is ¼ in. for every length equal to 40 diameters of the main pipe. As an example of this method of figuring suppose our main outlet is 48″ in diameter and that there are 10 sections proportioned as in the previous example. We will also say that the main section contains one elbow

and that there is also an elbow in the section 39″ in diameter, one elbow in the section 30″ in diameter, and another elbow in the section 20″ in diameter. Let the length of the pipe to the furthest outlet be 120 feet. We compute the friction then in the following way.

120 feet is equivalent to	30 diameters of 48″ pipe
1–48″ elbow is equivalent to	10 diameters of 48″ pipe
1–39″ elbow is equivalent to	10 diameters of 39″ pipe or 8.13 diameters of 48″ pipe
1–30″ elbow is equivalent to	10 diameters of 30″ pipe or 6.25 diameters of 48″ pipe
1–20″ elbow is equivalent to	10 diameters of 20″ pipe or 4.17 diameters of 48″ pipe

Total equivalent length 58.55 diameters of 48″ pipe.

The equivalent loss in velocity head will then be

$$58.55 \div 40 \quad 1.46$$

times the velocity head in the 48″ main. Further there is the velocity remaining in the 20″ pipe which gives an additional loss evidently of $^{20}/_{48}$ of one velocity head or .42 times the velocity head in the 48″ main. This gives a total loss in the piping system of

$$1.46 + .42 = 1.88$$

times the velocity head in the 48″ main. Assuming that the velocity in the 48″ main is 2000 feet per minute corresponding to a velocity head of ¼ in., the loss of pressure in the piping system is then

$$\tfrac{1}{4} \text{ in.} \times 1.88 = .47 \text{ in.}$$

This amount is to be deducted from the total pressure of the fan instead of from the static pressure when the piping is connected directly with the fan outlet, as by the reduction of velocity in the piping we have utilized practically all the velocity pressure at the fan outlet. In a "blow through" apparatus, however, this loss in pressure must be deducted from the static pressure; allowance must likewise be made for the loss in entrance to the piping which may be estimated at 45% of the velocity head. It will thus be seen that a "blow through" system requires larger piping than the "draw through" system for the same results.

In ordinary "draw through" heating system apparatus it is usually advisable to limit the pressure loss in piping to 50% of the total pressure. In the above example it has been shown that .47 in. out of the total pressure of 1 in. is lost if we make the pipe the same

size as the fan outlet, and therefore this is safe. However, if pressure loss had been .65 in. and we wished to reduce to .5 in. we could use the following formula, as loss in pressure varies approximately as the square of the velocity.

$$C_2 = C_1\sqrt{\frac{p_2}{p_1}} = C_1\sqrt{\frac{.50}{.65}} = .88C_1$$

Thus to get the same capacity with .5 in. loss as with .65 in. loss it would be necessary to increase the area of the piping throughout nearly $^{100}/_{88} = 13\%$, or the diameters of all the pipes approximately 6%. Then instead of a 48″ pipe it would be necessary to use a 51″ pipe, instead of a 46″ pipe a 49″ pipe, etc.

Heater Connection

Care should be taken to have the connection between the fan and the heater case of such a character that it will not restrict the flow of air or offer unnecessary resistance. This precaution is frequently overlooked, either throwing excessive pressure on the fan, or cutting down the quantity of air handled.

The following table gives the approximate lengths of connections advised for draw through installations.

LENGTH OF HEATER CONNECTION

For Draw Through Equipment.

SIZE FAN		DISTANCE FROM FAN TO HEATER
PLANOIDAL	NIAGARA CONOIDAL	
Up to 70″	Up to No. 7	18″—24″
70″—100″	No. 7—No. 10	24″—30″
100″—130″	No. 10—No. 13	36″
130″—170″	No. 13—No. 17	42″
170″—200″	No. 17—No. 20	48″—54″

Friction of Heaters

It is even more essential to take account of the friction of the air in passing through the heaters than through the piping. The loss of pressure here is much greater than ordinarily imagined and consequently many designers make the mistake of assuming higher velocities than are possible. The following table is compiled from careful tests on Buffalo Open Area Heaters and is perfectly reliable:

FRICTION OF AIR THROUGH REGULAR OPEN AREA PATTERN AND RETURN BEND BUFFALO HEATERS

LOSS OF AIR PRESSURE IN INCHES OF WATER PER SQUARE INCH—AIR AT 70° F

VELOCITY THROUGH CLEAR AREA	NUMBER OF SECTIONS							
	1	2	3	4	5	6	7	8
300	0.009	0.017	0.026	0.035	0.043	0.052	0.060	0.069
400	0.015	0.031	0.046	0.062	0.077	0.092	0.108	0.123
500	0.024	0.049	0.073	0.095	0.104	0.144	0.168	0.192
600	0.035	0.069	0.104	0.138	0.173	0.207	0.242	0.276
700	0.047	0.094	0.141	0.188	0.235	0.282	0.329	0.376
800	0.061	0.123	0.184	0.245	0.306	0.368	0.429	0.490
900	0.078	0.155	0.233	0.311	0.388	0.466	0.544	0.621
1000	0.096	0.191	0.287	0.382	0.479	0.574	0.670	0.765
1100	0.116	0.232	0.347	0.463	0.579	0.695	0.810	0.926
1200	0.138	0.276	0.414	0.551	0.689	0.827	0.965	1.103
1300	0.162	0.324	0.486	0.648	0.810	0.972	1.133	1.296
1400	0.187	0.375	0.562	0.750	0.936	1.124	1.311	1.500
1500	0.215	0.431	0.646	0.861	1.077	1.293	1.508	1.722
1600	0.245	0.490	0.735	0.980	1.226	1.471	1.716	1.961
1700	0.277	0.555	0.831	1.110	1.387	1.664	1.940	2.218
1800	0.310	0.620	0.930	1.240	1.550	1.860	2.167	2.480

The above losses are figured for air volumes at 70°. For accurate estimating, correction should be made for the increase in volume due to rise in temperature. The above factors enable us to read very readily the loss of pressure through the heaters. It is usually advisable to keep the loss in pressure in passing through the heaters down to 50% of the total pressure or less. Therefore for various pressures and various numbers of sections, the factors given in the following table and based on 50% pressure loss generally should not be exceeded.

MAXIMUM ALLOWABLE VELOCITIES OF AIR THROUGH CLEAR AREA OF HEATER FOR VARIOUS FAN PRESSURES AND FOR VARIOUS DEPTHS OF HEATER

	TOTAL FAN PRESSURE IN INCHES						
NUMBER OF SECTIONS DEEP	¾	1	1¼	1½	1¾	2	2½
4	990	1140	1280	1400	1510	1610	1800
5	885	1020	1140	1250	1350	1440	1610
6	810	930	1040	1140	1230	1320	1470
7	745	860	960	1055	1140	1220	1360
8	700	810	910	995	1070	1150	1280

The proper velocity for the air through the clear area of the heater will vary with the different conditions such as pressure carried and character of the installation. The table of velocities on page 111 is based on the assumption that the pressure loss through the heater should not exceed 50% of the total pressure on the fan.

The velocities here given are intended merely to indicate the practical limit, and except where the ducts are very short it will be found advisable to keep below this. This is especially true in the case of public buildings, where the limit should not exceed 90% of the above. The following table gives the maximum velocities advisable both for public buildings and for industrial plants for the different depths of heater indicated. These are based on the average pressures usually carried in such installations.

MAXIMUM VELOCITY ADVISABLE THROUGH HEATER FOR DIFFERENT INSTALLATIONS

Depth of Heater in Sections	In Public Buildings	In Industrial Plants
4	1140	1500
5	1020	1350
6	930	1230
7	860	1140
8	810	1070

UP DISCHARGE PLANOIDAL FAN DIRECT CONNECTED TO DOUBLE VERTICAL, DOUBLE-ACTING ENGINE

PROPERTIES OF DRY AIR

Barometric Pressure 29.921 Inches

Temperature Degrees Fahr.	Weight per Cu. Ft. Pounds	Per Cent of Volume at 70° F.	B. T. U. Absorbed by one Cu. Ft. Dry Air per degree F.	Cu. Ft. Dry Air Warmed one degree per B. T. U.	Temperature Degrees Fahr.	Weight per Cu. Ft. Pounds	Per Cent of Volume at 70° F.	B. T. U. Absorbed by one Cu. Ft. Dry Air per degree F.	Cu. Ft. Dry Air Warmed one degree per B.T.U.
0	.08636	.8680	.02080	48.08	130	.06732	1.1133	.01631	61.32
5	.08544	.8772	.02060	48.55	135	.06675	1.1230	.01618	61.81
10	.08453	.8867	.02039	49.05	140	.06620	1.1320	.01605	62.31
15	.08363	.8962	.02018	49.56	145	.06565	1.1417	.01592	62.82
20	.08276	.9057	.01998	50.05	150	.06510	1.1512	.01578	63.37
25	.08190	.9152	.01977	50.58	160	.06406	1.1700	.01554	64.35
30	.08107	.9246	.01957	51.10	170	.06304	1.1890	.01530	65.36
35	.08025	.9340	.01938	51.60	180	.06205	1.2080	.01506	66.40
40	.07945	.9434	.01919	52.11	190	.06110	1.2270	.01484	67.40
45	.07866	.9530	.01900	52.64	200	.06018	1.2455	.01462	68.41
50	.07788	.9624	.01881	53.17	220	.05840	1.2833	.01419	70.48
55	.07713	.9718	.01863	53.68	240	.05673	1.3212	.01380	72.46
60	.07640	.9811	.01846	54.18	260	.05516	1.3590	.01343	74.46
65	.07567	.9905	.01829	54.68	280	.05367	1.3967	.01308	76.46
70	.07495	1.0000	.01812	55.19	300	.05225	1.4345	.01274	78.50
75	.07424	1.0095	.01795	55.72	350	.04903	1.5288	.01197	83.55
80	.07356	1.0190	.01779	56.21	400	.04618	1.6230	.01130	88.50
85	.07289	1.0283	.01763	56.72	450	.04364	1.7177	.01070	93.46
90	.07222	1.0380	.01747	57.25	500	.04138	1.8113	.01018	98.24
95	.07157	1.0472	.01732	57.74	550	.03932	1.9060	.00967	103.42
100	.07093	1.0570	.01716	58.28	600	.03746	2.0010	.00923	108.35
105	.07030	1.0660	.01702	58.76	700	.03423	2.1900	.00847	118.07
110	.06968	1.0756	.01687	59.28	800	.03151	2.3785	.00782	127.88
115	.06908	1.0850	.01673	59.78	900	.02920	2.5670	.00728	137.37
120	.06848	1.0945	.01659	60.28	1000	.02720	2.7560	.00680	147.07
125	.06790	1.1040	.01645	60.79	1200	.02392	3.1335	.00603	165.83

PROPORTIONING DUCTS FOR PUBLIC BUILDINGS

In public buildings the sizes of air-conveying ducts from fans or heaters to vertical induction flues, and the sizes of these flues, depend upon the velocities of the air flowing in such ducts and flues. The essential factors in determining these velocities are: The limitations of economical rotative speed of fans from the standpoint of power, the limitations of air velocities on account of noise or by reason of increasing friction as velocities increase; limitation of velocity of inflowing air through registers into rooms; the desirability of as high a velocity of air as is permissible under the limitations referred to in order to get as quick a conveyance of heat units from the heater to the rooms to be heated as possible; and the necessary initial and intermediate velocities to overcome the resistance existing in each particular system or case.

The size of vertical flues to the registers in the rooms is determined by the maximum velocities allowable in avoiding drafts and noise in the rooms. Practice has shown that the best velocities for the registers should be from 200 to 400 feet per minute over the face of the register depending upon the size and location; floor registers from 125 to 175 feet. The velocity in the vertical flues leading to the registers should be from 400 to 750. The sizes of these vertical flues is determined largely by the size of register desirable. In general, the velocity in these risers should be low, in order to obtain as uniform a velocity as possible over the register area.

The velocity in the horizontal ducts leading from the apparatus to the vertical risers is determined chiefly by the resistance of the duct. In practice these velocities will vary anywhere from 700 feet to 1200 feet depending upon the size, length of the duct, number of elbows, etc. A designer with considerable experience may proportion these ducts so as to give very uniform distribution without going into any extended calculation. However, it is desirable to have a correct method as a basis. For the benefit of engineers and architects we give here the method employed by this company in the determination of duct velocities and sizes.

The principal losses in piping systems for public buildings are in the horizontal ducts where the velocity is the highest. The losses in these ducts depend upon the velocity, the size and length of duct and upon the number of elbows. There is also considerable loss in pressure as the air enters the duct. An ideal system should

take all these factors into consideration, and so proportion the velocities that the resistance would be practically equal in all ducts regardless of the length, etc.

The system which we employ accomplishes this in a practical manner and at the same time avoids any laborious calculation. For each duct a factor may be obtained by inspection in accordance with the following formula:

$$F = 2\tfrac{1}{2} + \frac{L}{4W} + \frac{N}{5}$$

This factor represents the loss by friction in terms of velocity head. The first term, 2½, is approximately the number of times the velocity head lost by entrance to the pipe, entrance to the vertical flue, and loss in riser and register. The second factor represents the loss due to length and size of pipe; L is the length in feet and W is the approximate width in inches. The third term represents that proportion of the pressure lost in elbows, and N is the number of long radius elbows. One square elbow is figured equal to two long radius elbows. In checking over the piping layout the factors for the various ducts are first found as above and from these factors the velocity in the respective ducts are ascertained directly. In determining these velocities it is usual to allow a loss not exceeding one-fourth of the total fan pressure. This in practice usually amounts to about ¼ of an inch. The velocity corresponding to a pressure of one-quarter of an inch is 2000, and since the velocities vary as the square root of the pressure, the factor F and the velocity V will give a loss of ¼ of an inch if

$$V = \frac{2000}{\sqrt{F}}$$

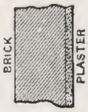
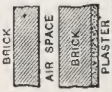
In this manner the velocities are accurately and conveniently proportioned. The table on page 117 from an actual case illustrates the variation in velocities which occur in a correctly proportioned system, and the table on page 119 shows standard size of registers and risers in public buildings.

Heat Losses in Buildings

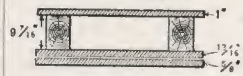
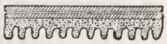
The table on page 116 shows the factors for heat losses for various forms of building construction as usually accepted by authorities on Heating and Ventilation, with a few changes made by our engineers after thorough research.

B. T. U. TRANSMITTED PER HOUR PER SQUARE FOOT SURFACE

For various differences in temperature

Kind of Surface	Difference in Temperature: Thickness	1°	50°	60°	70°
Brick wall bricks 8¼″x4″ x 2″ with vertical mortar joints ⅜″thick	8½″	.37	18.5	22.2	25.9
	13″	.29	14.5	17.4	20.3
BRICK 	17½″	.25	12.5	15.0	17.5
	22″	.22	11.0	13.2	15.4
	26½″	.19	8.5	11.4	13.3
Brick wall with ¾″ of plaster on one side	8½″	.36	18.0	21.6	25.2
	13″	.28	14.0	16.8	19.6
	17½″	.24	12.0	14.4	16.8
	22″	.21	10.5	12.6	14.7
	26½″	.18	9.0	10.8	12.6
Brick wall 2.4″ air space ¾″ plaster	8″	.25	12.5	15.0	17.5
	12½″	.21	10.5	12.6	14.7
	17″	.19	9.5	11.4	13.3
	21½″	.16	8.0	9.6	11.2
	26″	.14	7.0	8.4	9.8
Brick wall furred and ¾″ plaster	4″	.28	14.0	16.8	19.6
	8½″	.23	11.5	13.8	16.1
	13″	.20	10.0	12.0	14.0
	17½″	.18	9.0	10.8	12.6
	22″	.16	8.0	9.6	11.2
Frame walls  Clapboard 7/16 Studding Plaster		.44	22.0	26.4	30.8
Clapboard 7/16″ Paper Studding Plaster 		.31	15.5	18.6	21.7
 Clapboard 7/16″ Sheating ¾″ Studding Plaster		.28	14.0	16.8	19.6

Kind of Surface		Difference in Temperature: 1°	50°	60°	70°
Clapboard 7/16″ Paper Sheating ¾″ Studding Plaster 		.23	11.5	13.8	16.1
Single Window		1.09	54.5	65.4	76.3
Double Window		.46	23.0	27.6	32.2
Single Skylight		1.16	53.0	60.6	81.2
Double Skylight		.48	24.0	28.8	33.6
Doors or	1″	.41	20.5	24.6	28.7
wooden walls	1½″	.32	16.0	19.2	22.4
(Pine)	2″	.27	13.5	16.2	18.9

	1°
Single Flooring  For Ceiling	.10
For Floor	.07
Double Flooring  For Ceiling	.09
For Floor	.06
Corrugated Iron and Concrete with Double Flooring 	.13
Floor, single: no plaster beneath joists	.45
Floor, single ⅞″: lath and plaster beneath joists	.26
Floor, double 1½″: no plaster beneath joists	.31
Floor, double 1½″ lath and plaster beneath joists	.18
Ordinary stud partition: lath and plaster one side	.60
Ordinary stud partition: lath and plaster both sides	.34

Note:— Thicknesses of Brick Walls Refer to Actual Brick, Exclusive of Air Spaces, Plaster, Furring, Etc.

THE FOLLOWING TABLE FROM AN ACTUAL CASE ILLUSTRATES THE VARIATION IN VELOCITIES WHICH OCCUR IN A CORRECTLY PROPORTIONED SYSTEM

No. of Room	Cubic Feet Contents	Total B. T. U. Loss	A. P. M. Required for Heating	A. P. M. Required for Vent.	A. P. M. Allowed	Min. Air Change	A. P. M. for Each Duct	Factor	Velocity in Duct	Area of Duct Sq. Ft.
1	5290	13020	260	352	352	15	352	3	950	3.71
2	25700	50380	1008	2570	2570	10	1285	5	730	1.75
3	6070	36240	725	405	760	8	760	6	670	1.14
4	3530	14015	280	235	280	13	280	3	950	.3
5	1860	7985	159	93	159	12	160	3½	880	.19
6	3400	13255	265	227	265	13	265	5	730	.37
7	6070	30370	726	405	726	9	726	7	630	1.16
8	1860	7960	159	93	159	12	150	4	820	.19
9	55400	167000	3340	4440	4440	12½	2220	7	670	3.6

Heating Requirements of Buildings

Before deciding on the heating capacity required, the engineer of necessity makes up his estimate of the heat losses in the coldest weather. The principal loss is by radiation, and fortunately accurate data is available for use. The factors in the table on page 116 are subject to modifications to allow for exposure to winds, unequal distribution of heat, infiltration of cold air, etc. The German government standards call for these factors to be increased as follows:

Ten per cent. where the exposure is a northerly one and the winds are to be counted on as important factors.

Ten per cent. when the building is heated during the daytime only and the location of the building is *not* an exposed one.

Thirty per cent. when the building is heated during the daytime only, and the location of the building *is* exposed.

Fifty per cent. when the building is heated during the winter months intermittently, with long intervals of non-heating.

The heat required for ventilating is easily computed when the air supplied per hour is actually known. Since the specific heat of air at constant pressure is 0.238 and the weight of one cu. ft. of air at 70° is 0.075 lbs., one British Thermal Unit of heat will raise the temperature of one cu. ft. of air

$$\left\{\frac{1}{0.238 \times 0.0748}\right\} = 56^\circ\ F.$$

Infiltration Loss of heat through infiltration may properly be classed with ventilation losses. It varies greatly with the construction of the building and ranges from one air change in half an hour in a small and poorly constructed building, to one air change in two to three hours in a large well constructed building. This infiltration is caused in part by winds, but chiefly by the chimney-like effect of the column of air in a building at a higher temperature than that outside. The difference in pressure produced is proportional to the difference in temperature and the amount of infiltration is proportional to the square root of the difference in temperature, hence the heat losses due to infiltration may be expressed by the equation

$$H = c\ (t_2 - t_1)^{\frac{3}{2}}$$

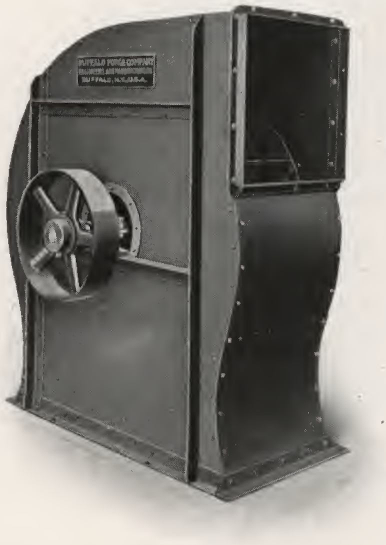
SIZE OF STEAM PIPES TO CARRY GIVEN HORSE POWER STEAM AT GIVEN VELOCITY

Steam at 5 Pounds Pressure, 20 Cubic Feet per Pound of Steam
Return Main Drop 1 Inch in 10 Feet

H. P. of 30 Pounds of Water	Velocity of Steam per Second					Return Main
	60	80	100	120	140	1
	Size of Pipe					
10	2½	2	2	2	1½	⅝
20	3½	3	2½	2½	2½	1
30	4	3½	3	3	2½	1¼
40	4½	4	3½	3½	3	1¼
50	5	4½	4	3½	3½	1½
60	6	5	4½	4	4	2
70	6	5	4½	4½	4	2
80	7	6	5	4½	4½	2
90	7	6	5	5	4½	2
100	7	6	6	5	5	2
120	8	7	6	6	5	2½
140	9	8	7	6	6	2½
160	9	8	7	6	6	2½
180	10	9	8	7	7	3
200	10	9	8	7	7	3
225	11	10	8	8	7	3
250	11	10	9	8	8	3½
275	12	10	9	9	8	3½
300	12	11	10	9	8	3½

STANDARD SIZES OF REGISTERS AND RISERS FOR PUBLIC BUILDINGS

Cu. Ft. of Air per Minute	Register Size Inches	Av. Vel. Over Face of Reg.	Size of Riser Inches	Riser Velocity Ft. per Minute
160	8 x 13	220	6 x 8	490
230	8 x 18	230	8 x 8	510
290	10 x 18	230	8 x 10	525
360	12 x 18	240	8 x 12	540
430	14 x 18	245	8 x 14	555
510	16 x 18	255	8 x 16	570
580	12 x 30	230	12 x 12	580
690	14 x 24	295	12 x 14	590
810	16 x 28	260	12 x 16	605
925	18 x 27	275	12 x 18	615
1040	20 x 26	290	12 x 20	625
1160	22 x 28	270	12 x 22	635
1290	24 x 27	285	12 x 24	645
1450	20 x 36	290	16 x 20	653
1620	22 x 36	295	16 x 22	663
1790	24 x 36	300	16 x 24	672
1970	24 x 36	330	16 x 26	680
2140	27 x 38	300	16 x 28	687
2310	30 x 36	310	16 x 30	693
2490	30 x 36	330	16 x 32	700



FULL HOUSING LEFT-HAND TOP HORIZONTAL DISCHARGE
PLANOIDAL FAN, WITH OVERHUNG PULLEY

TEMPERATURE OBTAINED WITH BUFFALO STANDARD HEATERS

GAUGE PRESSURE 5 LB., STEAM TEMP. 227° F.

VELOCITY OF AIR IN FT. PER MIN., MEASURED AT 70° F. AND 29.92″ BAROM.

Temp. of Entering Air	No. of Heater Sections	200		400		600		800		1000		1200		1400	
		Final Temp.	B. T. U. Per Lin. Ft. Per Hr.	Final Temp.	B. T. U. Per Lin. Ft. Per Hr.	Final Temp.	B. T. U. Per Lin. Ft. Per Hr.	Final Temp.	B. T. U. Per Lin. Ft. Per Hr.	Final Temp.	B. T. U. Per Lin. Ft. Per Hr.	Final Temp.	B. T. U. Per Lin. Ft. Per Hr.	Final Temp.	B. T. U. Per Lin. Ft. Per Hr.
-20°	1	23.0	261	17.9	459	14.1	620	11.1	754	8.9	876	6.9	976	5.1	1065
	2	58.0	236	49.5	421	43.1	574	38.0	703	34.1	820	30.2	913	27.2	1001
	3	87.1	216	76.5	390	68.4	536	61.5	659	56.1	769	51.1	862	46.9	946
	4	111.0	198	99.3	362	90.0	500	82.1	619	75.5	724	69.8	817	64.4	898
	5	131.2	183	118.4	336	108.5	467	100.0	582	92.7	683	86.1	772	80.6	854
	6	147.9	169	135.2	314	124.6	438	115.6	548	108.0	647	100.8	732	94.9	813
	7	161.5	157	149.0	293	138.5	412	129.3	517	121.3	612	114.0	696	107.8	775
	8	172.7	146	160.7	274	150.3	387	141.0	488	133.0	580	125.7	662	119.1	738
-10°	1	31.1	249	26.0	436	22.5	591	20.0	728	17.7	839	15.3	918	13.8	1010
	2	64.6	226	56.3	402	50.3	553	45.5	673	41.5	781	37.8	869	34.9	953
	3	92.6	207	82.4	374	74.8	514	68.1	631	62.8	736	57.8	822	53.9	904
	4	115.4	190	104.0	346	95.4	479	87.8	593	81.4	693	75.5	778	70.9	858
	5	135.0	176	122.7	322	113.0	447	105.0	558	97.9	654	91.4	738	86.1	816
	6	151.0	163	138.9	301	128.6	420	120.1	526	112.4	618	105.7	701	99.9	777
	7	164.0	151	152.0	281	141.9	395	133.0	495	125.2	585	118.3	667	112.3	741
	8	174.7	140	163.2	262	153.3	371	144.4	468	136.4	555	129.5	634	123.0	705
0°	1	39.5	239	34.5	418	31.1	566	28.5	691	26.4	800	24.2	878	22.6	959
	2	71.5	217	63.5	385	58.0	527	53.0	643	49.3	747	45.6	829	42.9	910
	3	98.2	198	88.4	357	81.0	491	74.6	603	69.5	702	65.0	788	61.0	863
	4	120.2	182	109.2	331	100.8	458	93.5	567	87.2	661	81.8	744	77.2	819
	5	138.9	168	127.3	309	117.9	429	110.0	534	103.0	624	97.0	706	91.9	780
	6	154.1	156	142.4	288	132.7	402	124.3	502	117.0	591	110.6	670	105.0	743
	7	166.6	144	155.0	268	145.4	378	136.8	474	129.2	559	122.6	637	116.7	708
	8	177.0	134	166.0	251	156.3	355	147.7	448	140.0	530	133.2	605	127.1	674

TEMPERATURE OBTAINED WITH BUFFALO STANDARD HEATERS
(CONTINUED)

Temp. of Entering Air	No. of Heater Sections	GAUGE PRESSURE 5 LB., STEAM TEMP. 227° F. VELOCITY OF AIR IN FT. PER MIN., MEASURED AT 70° F. AND 29.92" BAROM.													
		200		400		600		800		1000		1200		1400	
		Final Temp.	B.T.U. Per Lin. Ft. Per Hr.	Final Temp.	B.T.U. Per Lin. Ft. Per Hr.	Final Temp.	B.T.U. Per Lin. Ft. Per Hr.	Final Temp.	B.T.U. Per Lin. Ft. Per Hr.	Final Temp.	B.T.U. Per Lin. Ft. Per Hr.	Final Temp.	B.T.U Per Lin. Ft. Per Hr.	Final Temp.	B.T.U. Per Lin. Ft. Per Hr.
10°	1	47.6	228	43.0	400	39.5	537	37.1	637	35.0	758	33.1	838	31.5	913
	2	78.3	207	71.0	370	65.3	503	60.8	616	56.8	710	53.5	791	50.9	868
	3	103.7	189	94.5	342	87.4	469	81.2	576	76.2	669	71.9	751	68.0	821
	4	124.8	174	114.4	317	106.3	438	99.2	541	93.1	630	88.0	710	83.7	782
	5	142.7	161	131.7	295	122.7	410	115.0	509	108.3	596	102.6	674	97.7	745
	6	157.3	149	146.1	275	136.7	384	128.8	480	121.6	564	115.6	640	110.1	708
	7	169.2	138	158.8	258	148.9	361	140.5	452	133.2	534	127.0	608	121.2	674
	8	179.1	128	168.7	241	159.3	340	151.0	428	143.8	507	137.0	578	131.1	643
20°	1	55.6	216	51.5	382	48.3	515	45.8	626	43.6	716	41.9	795	40.4	866
	2	85.1	197	78.0	352	72.7	479	68.3	586	64.4	673	61.5	755	58.8	823
	3	109.4	181	100.5	326	94.0	449	88.0	550	82.8	635	79.0	716	75.3	783
	4	129.6	166	119.6	302	111.6	417	104.9	515	99.1	600	94.1	674	90.0	743
	5	146.7	154	135.6	280	127.4	391	120.0	485	113.4	566	108.1	641	103.4	708
	6	160.3	142	149.8	262	140.8	366	133.0	457	126.0	536	120.5	610	115.2	674
	7	171.8	132	161.2	245	152.4	344	144.4	431	137.3	508	131.4	579	125.9	642
	8	181.2	122	171.3	229	162.3	324	154.5	408	147.1	482	141.0	550	135.4	612
60°	1	88.6	173	85.4	308	82.9	417	80.5	497	79.0	576	77.6	639	76.3	692
	2	112.2	158	106.8	284	102.3	385	98.7	469	95.5	538	93.0	600	91.0	658
	3	132.2	146	125.0	263	119.0	358	114.4	440	110.4	509	107.0	570	104.2	625
	4	148.6	134	130.5	244	133.7	335	128.2	414	123.4	481	119.4	540	116.0	594
	5	162.0	124	153.2	226	146.1	313	140.1	389	134.8	454	130.4	512	126.4	564
	6	173.1	114	164.4	211	157.0	294	150.6	366	145.2	430	140.1	486	135.9	537
	7	182.4	106	174.0	198	166.5	277	160.0	347	154.2	408	149.0	463	144.6	513
	8	190.0	99	182.0	185	174.7	261	168.0	327	162.2	387	157.2	442	152.7	492

CAPACITIES OF BUFFALO PLANOIDAL STEEL PLATE EXHAUSTERS (TYPE L) UNDER AVERAGE WORKING CONDITIONS

TEMP. OF 70° F. 29.92″ BAROM.

Size	Diameter of Blast-Wheel	Area of Outlet Sq. Ft.	½″ Total Press. or 0.288 oz.			¾″ Total Press. or 0.433 oz.			1″ Total Press. or 0.577 oz.			1½″ Total Press. or 0.865 oz.		
			R.P.M.	VOL.	H. P.	R.P.M.	VOL.	H.P.	R.P.M.	VOL.	H.P.	R.P.M.	VOL.	H.P.
30	19¼″	0.77	620	1030	0.18	760	1260	0.32	877	1450	0.50	1074	1770	0.91
35	22½″	1.04	532	1400	0.24	651	1710	0.44	752	1970	0.68	921	2410	1.25
40	25¾″	1.36	465	1820	0.31	570	2230	0.58	658	2580	0.89	806	3150	1.63
45	29⅞″	1.75	414	2010	0.40	506	2820	0.73	585	3260	1.12	716	3990	2.06
50	32⅛″	2.16	372	2850	0.49	456	3490	0.90	526	4030	1.38	645	4930	2.54
55	35⅜″	2.61	338	3440	0.59	414	4220	1.09	478	4870	1.68	586	5960	3.08
60	38½″	3.13	310	4100	0.71	380	5020	1.30	439	5800	1.99	537	7100	3.66
70	45″	4.26	266	5580	0.96	326	6830	1.76	376	7890	2.71	460	9650	4.99
80	51⅜″	5.54	233	7290	1.25	285	8920	2.30	329	10300	3.54	403	12620	6.51
90	57⅞″	7.10	207	9220	1.59	253	11290	2.92	292	13040	4.49	358	15970	8.24
100	64¼″	8.75	186	11380	1.96	228	13940	3.60	263	16100	5.54	322	19720	10.20
110	70¾″	10.57	169	13770	2.37	207	16870	4.36	239	19480	6.70	293	23860	12.30
120	77¼″	13.00	155	16390	2.82	190	20080	5.18	219	23180	7.97	269	28390	14.70
130	83½″	14.85	143	19240	3.31	175	23560	6.08	202	27210	9.36	248	33320	17.20
140	90″	17.20	133	22310	3.84	163	27330	7.05	188	31560	10.90	230	38650	19.90
150	96½″	19.70	124	25610	4.41	152	31370	8.10	175	36230	12.50	215	44360	22.90
160	103″	22.40	116	29140	5.01	142	35690	9.21	164	41220	14.20	201	50470	26.00
170	109¼″	25.40	110	32900	5.66	134	40290	10.40	155	46530	16.00	190	56980	29.40
180	115¾″	28.50	103	36880	6.34	127	45170	11.70	146	52160	17.90	179	63880	33.00
190	122¼″	31.70	98	41100	7.07	120	50330	12.90	139	58120	20.00	170	71180	36.70
200	128½″	35.30	93	45540	7.83	114	55760	14.40	132	64400	22.20	161	78870	40.70

STATIC PRESSURE IS 79% OF TOTAL PRESSURE

CAPACITIES OF BUFFALO PLANOIDAL STEEL PLATE EXHAUSTERS (TYPE L) UNDER AVERAGE WORKING CONDITIONS

TEMP. OF 70° F. 29.92″ BAROM.

Size	Diameter of Blast-Wheel	Area of Outlet Sq. Ft.	2″ TOTAL PRESS. OR 1.154 OZ.			2½″ TOTAL PRESS. OR 1.442 OZ.			3″ TOTAL PRESS. OR 1.734 OZ.			3½″ TOTAL PRESS. OR 2.019 OZ.		
			R.P.M.	VOL.	H.P.	R.P.M.	VOL.	H.P.	R.P.M.	VOL.	H.P.	R.P.M.	VOL.	H.P.
30	19¼″	0.77	1240	2050	1.41	1387	2290	1.97	1519	2510	2.59	1641	2710	3.26
35	22½″	1.04	1064	2790	1.92	1189	3120	2.68	1302	3420	3.52	1406	3690	4.44
40	25¾″	1.36	930	3630	2.51	1040	4070	3.50	1139	4460	4.60	1230	4820	5.80
45	29⅞″	1.75	827	4610	3.17	924	5160	4.43	1013	5650	5.83	1094	6100	7.34
50	32⅛″	2.16	744	5690	3.92	832	6360	5.47	912	6970	7.19	984	7530	9.06
55	35⅜″	2.61	676	6890	4.74	756	7700	6.62	829	8440	8.70	895	9110	11.00
60	38½″	3.13	620	8200	5.64	693	9160	7.88	760	10040	10.40	820	10840	13.10
70	45″	4.26	532	11540	7.67	594	12470	10.70	651	13660	14.10	703	14760	17.80
80	51⅜″	5.54	465	14570	10.00	520	16290	14.00	570	17850	18.40	615	19280	23.20
90	57⅞″	7.10	413	18440	12.70	462	20600	17.70	506	22590	23.30	547	24400	29.40
100	64¼″	8.75	372	22770	15.70	416	25460	21.90	456	27900	28.80	492	30120	36.30
110	70¾″	10.57	338	27540	19.00	378	30800	26.50	414	33740	34.80	448	36450	43.90
120	77¼″	13.00	310	32780	22.60	347	36660	31.50	380	40650	41.40	410	43380	52.20
130	83½″	14.85	286	38470	26.50	320	43020	37.00	351	47100	48.60	379	50900	61.30
140	90″	17.20	266	44630	30.70	297	49890	42.90	326	54150	56.40	352	59040	71.00
150	96½″	19.70	248	51220	35.30	277	57260	49.30	304	62740	64.80	328	67770	81.60
160	103″	22.40	233	58270	40.10	260	65170	56.00	285	71370	73.60	308	77110	92.80
170	109¼″	25.40	219	65790	45.30	245	73570	63.30	268	80590	83.10	290	87060	104.80
180	115¾″	28.50	207	73760	50.70	231	82480	70.90	253	90340	93.20	274	97600	117.50
190	122¼″	31.70	196	82180	56.50	219	91900	79.00	240	100670	103.40	259	108740	130.90
200	128½″	35.30	186	91060	62.70	208	101800	87.60	228	111540	115.10	246	120490	145.00

STATIC PRESSURE IS 79% OF TOTAL PRESSURE

CAPACITIES OF BUFFALO NIAGARA CONOIDAL FANS (TYPE N) UNDER AVERAGE WORKING CONDITIONS

AT 70° F. AND 29.92″ BAROM.

Fan No.	Diam. of Blast-Wheel	Area of Outlet Sq. Ft.	½″ Total Press. or 0.288 Oz.			¾″ Total Press. or 0.433 Oz.			1″ Total Press. or 0.577 Oz.			1 ½″ Total Press. or 0.865 Oz.		
			R. P. M.	VOL.	H. P.	R. P. M.	VOL.	H. P.	R. P. M.	VOL.	H. P.	R. P. M.	VOL.	H. P.
3	15⅝″	1.31	478	1720	0.19	585	2110	0.35	675	2440	0.54	827	2990	1.00
3½	18⅛″	1.79	409	2350	0.26	501	2870	0.48	579	3320	0.74	709	4060	1.37
4	20½″	2.33	358	3070	0.34	439	3750	0.63	506	4340	0.97	620	5310	1.78
4½	23½″	2.95	318	3880	0.43	390	4750	0.80	450	5490	1.22	551	6720	2.25
5	26⅛″	3.64	287	4790	0.53	351	5870	0.98	405	6770	1.51	496	8300	2.77
5½	28¾″	4.41	260	5800	0.65	319	7100	1.19	368	8200	1.83	451	10040	3.36
6	31⅜″	5.25	239	6900	0.77	292	8450	1.41	338	9750	2.17	414	11940	4.00
7	36½″	7.14	205	9400	1.05	251	11500	1.92	289	13280	2.96	354	16260	5.44
8	42″	9.33	179	12260	1.37	219	15020	2.51	253	17340	3.87	310	21240	7.10
9	47″	11.81	159	15520	1.73	195	19000	3.18	225	21950	4.89	276	26880	8.99
10	52″	14.58	143	19160	2.14	175	23460	3.93	203	27090	6.04	248	33180	11.10
11	58″	17.64	130	23180	2.58	160	28390	4.75	184	32780	7.31	226	40150	13.40
12	63″	21.00	119	27590	3.08	146	33780	5.65	169	39010	8.70	207	47770	16.00
13	68″	24.65	110	32370	3.61	135	39650	6.63	156	45780	10.20	191	56070	18.80
14	73″	28.68	102	37550	4.19	125	45990	7.69	145	53100	11.80	177	65030	21.80
15	78″	32.80	96	43100	4.80	117	52790	8.83	135	60960	13.60	165	74650	25.00
16	84″	37.32	90	49040	5.47	110	60060	10.10	127	69360	15.50	155	84940	28.40
17	89″	42.14	84	55370	6.17	103	67800	11.40	119	78300	17.50	146	95900	32.10
18	94″	47.24	80	62060	6.92	98	76010	12.70	113	87780	19.60	138	107500	36.00
19	99″	52.63	75	69160	7.71	92	84700	14.20	107	97800	21.80	131	119780	40.10
20	105″	58.32	72	76640	8.54	88	93850	15.70	101	108370	24.20	124	132710	44.40

Static Pressure is 77½% of Total Pressure

CAPACITIES OF BUFFALO NIAGARA CONOIDAL FANS (TYPE N) UNDER AVERAGE WORKING CONDITIONS

AT 70° F. AND 29.92″ BAROM.

Fan No.	Diam. of Blast-Wheel	Area of Outlet Sq. Ft.	2″ Total Press. or 1.154 oz.			2½″ Total Press. or 1.442 oz.			3″ Total Press. or 1.734 oz.			3½″ Total Press. or 2.019 oz.		
			R. P. M.	VOL.	H. P.	R. P. M.	VOL.	H. P.	R. P. M.	VOL.	H. P.	R. P. M.	VOL.	H. P.
3	15⅝″	1.31	955	3450	1.54	1067	3860	2.15	1169	4220	2.83	1263	4560	3.56
3½	18⅛″	1.79	818	4690	2.09	915	5250	2.93	1002	5750	3.85	1083	6210	4.85
4	20½″	2.33	716	6130	2.73	801	6850	3.82	877	7510	5.02	947	8110	6.32
4½	23½″	2.95	636	7760	3.46	712	8670	4.83	780	9500	6.36	842	10260	8.01
5	26⅛″	3.64	573	9580	4.27	640	10710	5.96	702	11730	7.84	758	12670	9.87
5½	28¾″	4.41	521	11590	5.17	582	12960	7.22	638	14190	9.49	689	15330	12.00
6	31⅜″	5.25	477	13790	6.15	534	15420	8.59	585	16890	11.30	632	18250	14.20
7	36½″	7.14	409	18770	8.37	458	20990	11.70	501	23000	15.40	541	24840	19.40
8	42″	9.33	358	24520	10.90	400	27410	15.30	439	30040	20.10	474	32440	25.30
9	47″	11.81	318	31020	13.80	356	34700	19.30	390	38010	25.40	421	41050	32.00
10	52″	14.58	286	38310	17.10	320	42840	23.90	351	46930	31.40	379	50700	39.50
11	58″	17.64	260	46360	20.70	291	51800	28.90	319	56780	38.00	344	61330	47.80
12	63″	21.00	239	55170	24.60	267	61680	34.40	292	67570	45.20	316	72990	57.00
13	68″	24.65	220	64730	28.90	246	72380	40.30	270	79300	53.00	292	85650	66.80
14	73″	28.68	205	75090	33.50	229	83950	46.80	251	91970	61.50	271	99340	77.50
15	78″	32.80	191	86200	38.40	214	96380	53.70	234	105580	70.60	253	114050	89.00
16	84″	37.32	179	98060	43.70	200	109660	61.10	219	120130	80.30	237	129750	101.20
17	89″	42.14	169	110720	49.40	188	123800	69.00	206	135620	90.70	223	146490	114.30
18	94″	47.24	159	124110	55.30	178	138770	77.30	195	152020	101.70	211	164110	128.10
19	99″	52.63	151	138280	61.70	169	154620	86.20	185	169400	113.30	200	182970	142.70
20	105″	58.32	143	153250	68.30	160	171320	95.50	175	187680	125.50	190	202720	158.10

Static Pressure is 77½% of Total Pressure

PLANOIDAL EXHAUSTERS

With Proper Combinations of Heaters and Engines for Public Buildings and Industrial Installations.

Size of Fan	Cubic Feet of Air per Min.		Buffalo Standard Heater				Engine Size	
	1″ Total Press.	2″ Total Press.	Arrangement	Style	Size	Clear Area Sq. Ft.	Low Press. Steam	High Press. Steam
50	4030	5690	Single	R.O.A.	3′–0″ x 3′– 4″	4.4		
55	4870	6890	Single	R.O.A.	3′–0″ x 3′–10″	5.2		3x3½ I
					3′–0″ x 4′– 4″	6.0		
60	5800	8200	Single	R.O.A.	3′–0″ x 4′– 4″	6.0		
					3′–0″ x 4′–10″	6.8	8x6	4x3½ I
					3′–0″ x 5′– 4″	7.6		
70	7890	11540	Single	R.O.A.	3′–0″ x 5′– 4″	7.6		
					3′–0″ x 5′–10″	8.4	8x6	4x3½ I
					4′–0″ x 5′– 4″	9.7		
80	10300	14570	Single	R.O.A.	4′–0″ x 5′–10″	10.7		
					4′–0″ x 6′– 4″	11.2		
					4′–0″ x 6′–10″	12.6	8x6	5 x 5
					4′–6″ x 5′–10″	12.1		4½x5 I
					4′–6″ x 6′– 4″	13.1		
90	13040	18440	Single	R.O.A.	4′–6″ x 6′– 4″	13.1		
					4′–6″ x 6′–10″	14.2		
					4′–6″ x 7′– 4″	15.3	8x6	5 x5
					5′–0″ x 6′– 4″	14.1		4½x5 I
					5′–0″ x 6′–10″	15.4		
					5′–0″ x 7′– 4″	16.6		
100	16100	22770	Single	R.O.A.	5′–0″ x 6′–10″	15.4		6 x6
					5′–0″ x 7′– 4″	16.6	10x8	4½x5 I
					5′–0″ x 7′–10″	17.7		5x10 N
					6′–0″ x 7′– 4″	19.8		
110	19480	27540	Single	R.O.A.	6′–0″ x 7′– 4″	19.8		7 x7
					6′–0″ x 7′–10″	21.3		6 x8
					6′–0″ x 8′– 4″	22.7	10x8	6½x8 I
					6′–0″ x 8′–10″	24.2		5x10 N
				R.B.	7′–0″ x 7′– 4″	23.6		
120	23180	32780	Single	R.O.A.	6′–0″ x 8′– 4″	22.7		
					6′–0″ x 8′–10″	24.2		7 x7
				R. B.	7′–0″ x 7′– 4″	23.6	12x8	8 x8
					7′–0″ x 7′–10″	25.4		6½x8 I
					7′–0″ x 8′– 4″	27.2		6x10 N
					7′–0″ x 8′–10″	29.0		
130	27210	38470	Single	R.B.	7′–0″ x 8′– 4″	27.2		
					7′–0″ x 8′–10″	29.0		8 x8
					7′–0″ x 9′– 4″	30.7	12x8	6½x8 I
					7′–0″ x 9′–10″	32.5		6x10 N
					8′–6″ x 8′– 4″	33.2		
140	31560	44630	Single	R.B.	7′–0″x 9′– 4″	30.7		
					7′–0″x 9′–10″	32.5		
					8′–6″x 8′– 4″	33.2		
					8′–6″x 8′–10″	35.3		8 x8
					8′–6″x 9′– 4″	37.6	15x8	8 x10
					8′–6″x 9′–10″	39.8		7½x9 I
					8′–6″x 10′– 4″	41.8		7 x12N
					8′–6″x 10′–10″	44.0		
					9′–6″x 8′– 4″	36.7		
					9′–6″x 8′–10″	39.0		

PLANOIDAL EXHAUSTERS

With Proper Combinations of Heaters and Engines for Public Buildings and Industrial Installations

Size of Fan	Cubic Feet of Air per Minute		Buffalo Standard Heater				Engine Size	
	1″ Total Press.	2″ Total Press.	Arrangement	Style	Size	Clear Area Sq. Ft.	Low Press. Steam	High Press. Steam
150	36230	51220	Single	R.B.	8′–6″x 8′–10″	35.3	15x8	10x8 8x10 7½x9 I 7x12 N
					8′–6″x 9′– 4″	37.6		
					8′–6″x 9′–10″	39.8		
					8′–6″x10′– 4″	41.8		
					8′–6″x10′–10″	44.0		
					9′–6″x 8′– 4″	36.7		
					9′–6″x 8′–10″	39.0		
					9′–6″x 9′– 4″	41.4		
					9′–6″x 9′–10″	43.8		
					9′–6″x10′– 4″	46.0		
160	41220	58270	Single	R.B.	8′–6″x 9′–10″	39.8	15x10	10x10 8x12 N
					8′–6″x10′– 4″	41.8		
					8′–6″x10′–10″	44.0		
					9′–6″x 9′– 4″	41.4		
					9′–6″x 9′–10″	43.8		
					9′–6″x10′– 4″	46.0		
					9′–6″x10′–10″	48.4		
					9′–6″x11′– 4″	50.8		
					9′–6″x11′–10″	53.2		
			Back to Back	R.O.A.	6′–0″x 7′– 4″	39.6		
					6′–0″x 7′–10″	41.6		
					6′–0″x 8′– 4″	45.4		
					6′–0″x 8′–10″	48.4		
				R.B.	7′–0″x 7′– 4″	47.2		
					7′–0″x 7′–10″	50.8		
170	46530	65790	Single	R.B.	9′–6″x10′– 4″	46.0	15x10	10x10 8x14 N 9x14 N
					9′–6″x10′–10″	48.4		
					9′–6″x11′– 4″	50.8		
					9′–6″x11′–10″	53.2		
			Back to Back	R.O.A.	6′–0″x 8′– 4″	45.4		
					6′–0″x 8′–10″	48.4		
				R.B.	7′–0″x 7′– 4″	47.2		
					7′–0″x 7′–10″	50.8		
					7′–0″x 8′– 4″	54.4		
					7′–0″x 8′–10″	58.0		
					7′–0″x 9′– 4″	61.4		
180	52160	73760	Back to Back	R.B.	7′–0″x 7′–10″	50.8	16x10	12x10 10x12 9x14 N
					7′–0″x 8′– 4″	54.4		
					7′–0″x 8′–10″	58.0		
					7′–0″x 9′– 4″	61.4		
					7′–0″x 9′–10″	65.0		
					8′–6″x 8′– 4″	66.4		
190	58120	82180	Back to Back	R.B.	7′–0″x 8′–10″	58.0	18x12	12x12 9x14 N
					7′–0″x 9′– 4″	61.4		
					7′–0″x 9′–10″	65.0		
					8′–6″x 8′– 4″	66.4		
					8′–6″x 8′–10″	70.6		
					8′–6″x 9′– 4″	73.2		
					9′–6″x 8′– 4″	73.4		

NIAGARA CONOIDAL FANS

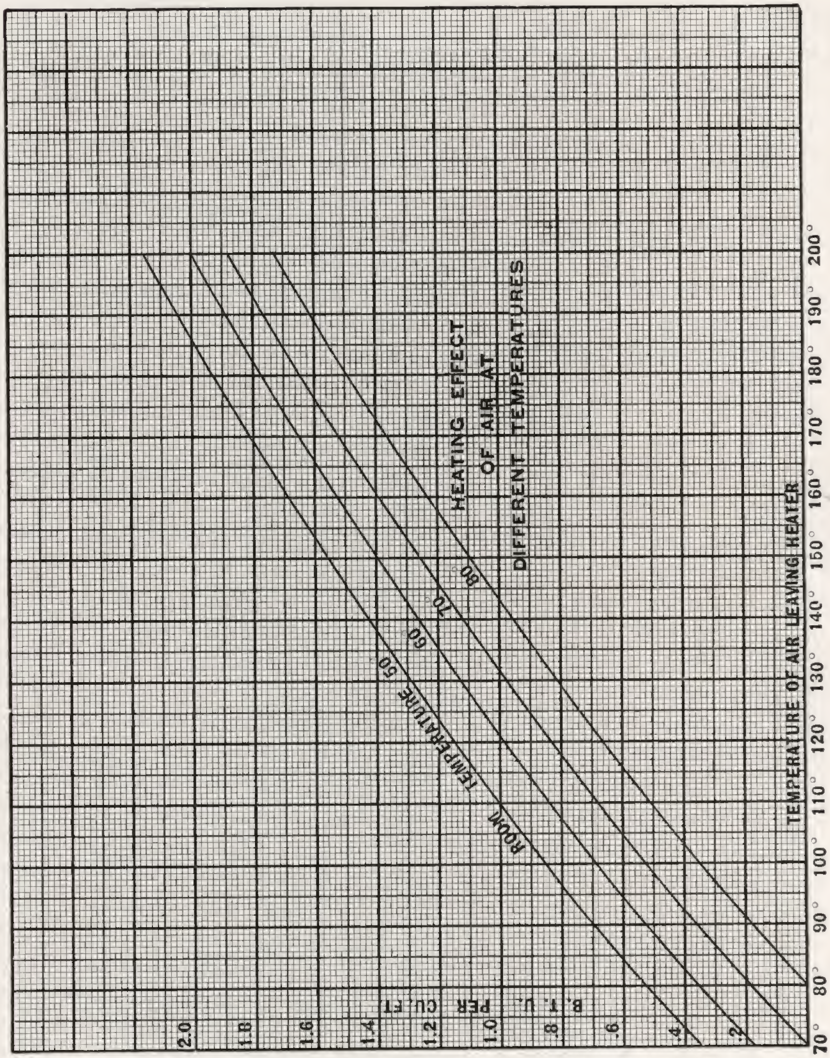
With Proper Combinations of Heaters and Engines for Public Buildings and Industrial Installations.

Fan No.	Cubic Feet of Air per Min.		Buffalo Standard Heater				Engine Size	
	1″Total Press.	2″Total Press.	Arrangement	Style	Size	Clear Area Sq. Ft.	Low Press. Steam	High Press. Steam
4	4340	6130	Single	R.O.A.	3′–0″ x 3′– 4″	4.4		
					3′–0″ x 3′–10″	5.2		
4½	5490	7760	Single	R.O.A.	3′–0″ x 3′–10″	5.2		4 x4 A 3 x3½ I
					3′–0″ x 4′– 4″	6.0		
					3′–0″ x 4′–10″	6.8		
5	6770	9580	Single	R.O.A.	3′–0″ x 4′– 4″	6.0	5x5	4 x4 A 4 x3½ I
					3′–0″ x 4′–10″	6.8		
					3′–0″ x 5′– 4″	7.6		
					3′–0″ x 5′–10″	8.4		
5½	8200	11590	Single	R.O.A.	3′–0″ x 5′– 4″	7.6	6x6	4 x4 A 4 x3½ I
					3′–0″ x 5′–10″	8.4		
					4′–0″ x 5′– 4″	9.7		
					4′–0″ x 5′–10″	10.7		
					4′–0″ x 6′– 4″	11.2		
6	9750	13790	Single	R.O.A.	4′–0″ x 5′– 4″	9.7	6x6	5 x5 A 4½x5 A
					4′–0″ x 5′–10″	10.7		
					4′–0″ x 6′– 4″	11.2		
					4′–0″ x 6′–10″	12.6		
					4′–6″ x 5′–10″	12.1		
					4′–6″ x 6′– 4″	13.1		
7	13280	18770	Single	R.O.A.	4′–0″ x 6′–10″	12.6	8x6	5 x5 A 5½x7 I
					4′–6″ x 5′–10″	12.1		
					4′–6″ x 6′– 4″	13.1		
					4′–6″ x 6′–10″	14.2		
					4′–6″ x 7′– 4″	15.3		
					5′–0″ x 6′– 4″	14.1		
					5′–0″ x 6′–10″	15.4		
					5′–0″ x 7′– 4″	16.6		
					5′–0″ x 7′–10″	17.7		
8	17340	24520	Single	R.O.A.	5′–0″ x 7′– 4″	16.6	10x8	6 x6 A 5½x7 I
					5′–0″ x 7′–10″	17.7		
					6′–0″ x 7′– 4″	19.8		
					6′–0″ x 7′–10″	21.3		
					6′–0″ x 8′– 4″	22.7		
9	21950	31020	Single	R.O A.	6′–0″ x 7′– 4″	19.8	10x8	6 x6 A 6½x8 I
					6′–0″ x 7′–10″	21.3		
					6′–0″ x 8′– 4″	22.7		
					6′–0″ x 8′–10″	24.2		
				R.B.	7′–0″ x 7′– 4″	23.6		
					7′–0″ x 7′–10″	25.4		
					7′–0″ x 8′– 4″	27.2		
10	27090	38310	Single	R.B	7′–0″ x 7′–10″	25.4	12x8	7 x7 A 6½x8 I 6 x10 N
					7′–0″ x 8′– 4″	27.2		
					7′–0″ x 8′–10″	29.0		
					7′–0″ x 9′– 4″	30.7		
					7′–0″ x 9′–10″	32.5		
					8′–6″ x 8′– 4″	33.2		
					8′–6″ x 8′–10″	35.3		

NIAGARA CONOIDAL FANS

With Proper Combinations of Heaters and Engines for Public Buildings and Industrial Installations.

Fan No.	Cubic Feet of Air per Min. 1″ Total Press.	2″ Total Press.	Buffalo Standard Heater Arrangement	Style	Size	Clear Area Sq. Ft.	Engine Size Low Press. Steam	High Press. Steam
11	32780	46360	Single	R. B.	7′–0″x 9′– 4″	30.7	12x 8	8x8 A 7½x9 I 7x12 N
					7′–0″x 9′–10″	32.5		
					8′–6″x 8′– 4″	33.2		
					8′–6″x 8′–10″	35.3		
					8′–6″x 9′– 4″	37.6		
					8′–6″x 9′–10″	39.8		
					8′–6″x10′– 4″	41.8		
					8′–6″x10′–10″	44.0		
					9′–6″x 8′– 4″	36.7		
					9′–6″x 8′–10″	39.0		
					9′–6″x 9′– 4″	41.4		
					9′–6″x 9′–10″	43.8		
12	39010	55170	Single	R.B.	8′–6″x 8′–10″	35.3	15x 8	10x 8 A 8x10 A 7x12 N
					8′–6″x 9′– 4″	37.6		
					8′–6″x 9′–10″	39.8		
					8′–6″x10′– 4″	41.8		
					8′–5″x10′–10″	44.0		
					9′–6″x 8′– 4″	36.7		
					9′–6″x 8′–10″	39.0		
					9′–6″x 9′– 4″	41.4		
					9′–6″x 9′–10″	43.8		
					9′–6″x10′– 4″	46.0		
					9′–6″x10′–10″	48.4		
					9′–6″x11′– 4″	50.8		
13	45780	64730	Single	R.B.	8′–6″x10′– 4″	41.8	15x 8	10x 8 A 8x12 N
					8′–6″x10′–10″	44.0		
					9′–6″x 9′– 4″	41.4		
					9′–6″x 9′–10″	43.8		
					9′–6″x10′– 4″	46.0		
					9′–6″x10′–10″	48.4		
					9′–6″x11′– 4″	50.8		
					9′–6″x11′–10″	53.2		
			Back to Back	R.O A.	6′–0″x 7′–10″	42.6		
					6′–0″x 8′– 4″	45.4		
					6′–0″x 8′–10″	48.4		
				R.B.	7′–0″x 7′– 4″	47.2		
					7′–0″x 7′–10″	50.8		
					7′–0″x 8′– 4″	54.4		
					7′–0″x 8′–10″	58.0		
14	53100	75090	Single	R.B.	9′–6″x10′–10″	48.4	15x10	10x10 A 8x14 N
					9′–6″x11′– 4″	50.8		
					9′–6″x11′–10″	53.2		
			Back to Back	R.O.A.	6′–0″x 8′–10″	48.4		
				R.B.	7′–0″x 7′–10″	50.8		
					7′–0″x 8′– 4″	54.4		
					7′–0″x 8′–10″	58.0		
					7′–0″x 9′– 4″	61.4		
					7′–0″x 9′–10″	65.0		
					8′–6″x 8′– 4″	66.4		
					8′–6″x 8′–10″	70.6		


HEATING EFFECT
OF AIR AT
DIFFERENT TEMPERATURES
TEMPERATURE OF AIR LEAVING HEATER
70° 80° 90° 100° 110° 120° 130° 140° 150° 160° 170° 180° 190° 200°
B. T. U. PER CU. FT.
.2 .4 .6 .8 1.0 1.2 1.4 1.6 1.8 2.0
ROOM TEMPERATURE 50 60 70 80

Heater Performance

In modern methods of determining the size of apparatus, whether for heating or drying, the heat losses are first calculated in the manner described on page 116. In public buildings the amount of air is usually specified and the required temperature of air for heating may be determined from the equation.

$$t_2 = \frac{k\,L}{0.238 \times 60 \times w\ A} + t_r$$

in which t_2 = the temperature of the air leaving the heater.

L = the B. T. U. per hour lost by transmission through walls, glass surfaces, roofs, etc.

A = the cu. ft. of air required for ventilation.

t_r = the temperature of the room.

w = weight of 1 cu. ft. of air (which taken at a temperature of 70°, is 0.075 lbs.)

k is an assumed factor of safety chosen with reference to the particular conditions.

This formula may also be used in determining the volume of air required when the temperature of the air is specified.

Where "air return" is used, the formula is modified as follows to give the total heat units required with view of choosing a standard size of apparatus to meet the conditions.

$$H = 0.238\ w\ n\ C\ (t_r - t_1) + k\ L$$

in which n = number of air changes per hour due to the infiltration of cold air from without. This is dependent upon the size and construction of the building and must be chosen as a result of experiments and tests upon various types of buildings.

C = the cu. ft. contents of the room.

t_1 = the outside temperature.

Heating Surface

The next step is to determine the total amount of heating surface in lineal feet of one inch pipe, and as the efficiency of the heating surface varies under different conditions, we select the proper velocity of flow of air per square foot of open area in one section (see bottom of page 111), as well as the number of four-row sections (usually five or six), and from the tables on pages 120 and 121 find the B. T. U. per lineal foot. This, divided into the total B. T. U. required gives the heating surface.

Size of Sections The size of the section is, of course, the result obtained by dividing the total heating surface by the number of sections, one section always being considered as an entire bank of coils four rows of pipe deep, whether arranged in *one group or more.*

Air Required Per Minute The cubic feet of air to be furnished is the product of the square feet clear area of one section and the velocity in feet per minute first assumed. A fan is then selected which has the proper capacity at a speed suited to the nature of the installation.

In the table on pages 120 and 121, five pounds pressure has been chosen as most serviceable, and for any other pressure correction may be made from the curves on page 136. These tables give final temperatures, and B. T. U. transmission for various numbers of sections in depth of Buffalo Heaters, and for various velocities of flow through the clear area of the heater.

It should be especially remembered that the values given in this table hold *only* for *Standard Open Area and Return Bend Buffalo Heaters.* Other standards, owing to variation in spacing and therefore variation in velocities and capacities, give *quite different* results. In using these tables the velocity through the heater selected depends largely on the number of sections used and the pressure at which the fan is run. Care should always be taken that the resistance is not too great for the fan pressure. As a guide to engineers in selecting proper velocities we give the table on page 112 which may be used in public buildings and in manufacturing buildings respectively with a various number of heater sections. Lower velocities may, of course, be used if preferred.

NIAGARA CONOIDAL FAN

APPLICATION OF HEATING TABLES

CASE I—HEATING FRAME MACHINE SHOP 60° F AT 0° F

Surface	B. T. U. Loss per Degree Diff.	Cubic Contents
Floor 24500 x .1	2450	671000 cu. ft.
Roof (2″ plank, tar and gravel) 27800 x .3 ...	8350	
Glass 11950 x 1.1 .	13150	
Wall (frame) 7950 x .4.........	3180	
Infiltration $\frac{671000}{56}$ (1 hr. air change)	12050	
Total B. T. U. loss per degree difference......	39180 B. T. U.	

These tables are applied in two ways. In the first case where air is returned to the apparatus from the building, there is no requirement for ventilation and the amount of heater necessary is determined by dividing the total B. T. U. required to heat the building by the B. T. U. transmitted per lineal foot of pipe, under the assumed conditions as shown by the accompanying table. The following is an illustration of a practical application of these methods.

39180 x 60 = 2350800 B. T. U. Total

Add 15% margin = 2700000 B. T. U.

Apparatus: Use 6 sec. 1200 ft. per minute velocity through clear area of Heater. Steam pressure 5 pounds and air returned at 60° from the building.

Then B. T. U. per lineal foot = 486 (page 121)

$\frac{2700000}{486}$ = 5550 total lineal feet necessary in heater.

$\frac{5550}{6}$ = 925 lineal feet per section.

Heater—6 sections 7′ x 7′10″ (4 rows) with 25.4 sq. ft. clear area (see page 79).

If heat loss has been figured close, use the next larger size heater.

A. P. M. = 25.4 x 1200 = 30500 cu. ft. at 70° F. or 34600 cu. ft. at 140° F. (Final temp.)

Fan: Assume 2″ total pressure.

If a steel plate fan is wanted refer to pages 122 and 123. Use 130″ Planoidal Type L, 258 R. P. M. If a multivane fan is to be used refer to pages 124 and 125. Use No. 10 Niagara Conoidal Type N, 260 R. P. M.

CASE II

70° F. at 0° F.	B. T. U. Loss per Degree Diff.	Cubic Contents
Roof (2″ board, tar, paper and gravel) 10310 x .26.	2680	206000 cu. ft.
Floor 9160 x .15 . .	1370	
Glass 720 x 1.1 . . .	790	
Walls (12″ Brick) 5800 x .34 . .	1970	

Total loss per degree difference 6810

In this application it is assumed that all air is taken from outdoors for the purpose of ventilation, and either the volume of air to be supplied is specified as in the illustration below and the final temperature is to be determined or the final temperature may be selected from the tables and the volume of air determined. The following is an illustration of the application in heating a building where a ten minute air change is specified.

6810 x 70 = 476700 B. T. U. loss + 25% = 595000 total loss.

Assume 10 minute air change. Then $\frac{206000}{10} = 20600$ A. P.

M. at 70° $(T-70°)\frac{20600 \times 60°}{56} \times 595000$ B. T. U.

$T—70° = 27.1°$

Final temp. $= T = 97.1°$

Heater 5 pounds steam pressure, 1000 feet velocity through clear area, 5 sections deep

$\frac{20600}{1000} = 20.6$ sq. ft. clear area through heater.

Use 5 sections 6′ x 7′10″ heater

763 x 5 = 3815 total lineal feet of heater (See page 79)

Fan: Assume 1″ total pressure.

If a steel plate fan is to be used refer to pages 122 and 123. Use 120″ Planoidal Type "L," 195 R. P. M. If a multivane fan is wanted refer to pages 124 and 125. Use No. 9 Niagara Conoidal Type N, 212 R. P. M.

CASE III

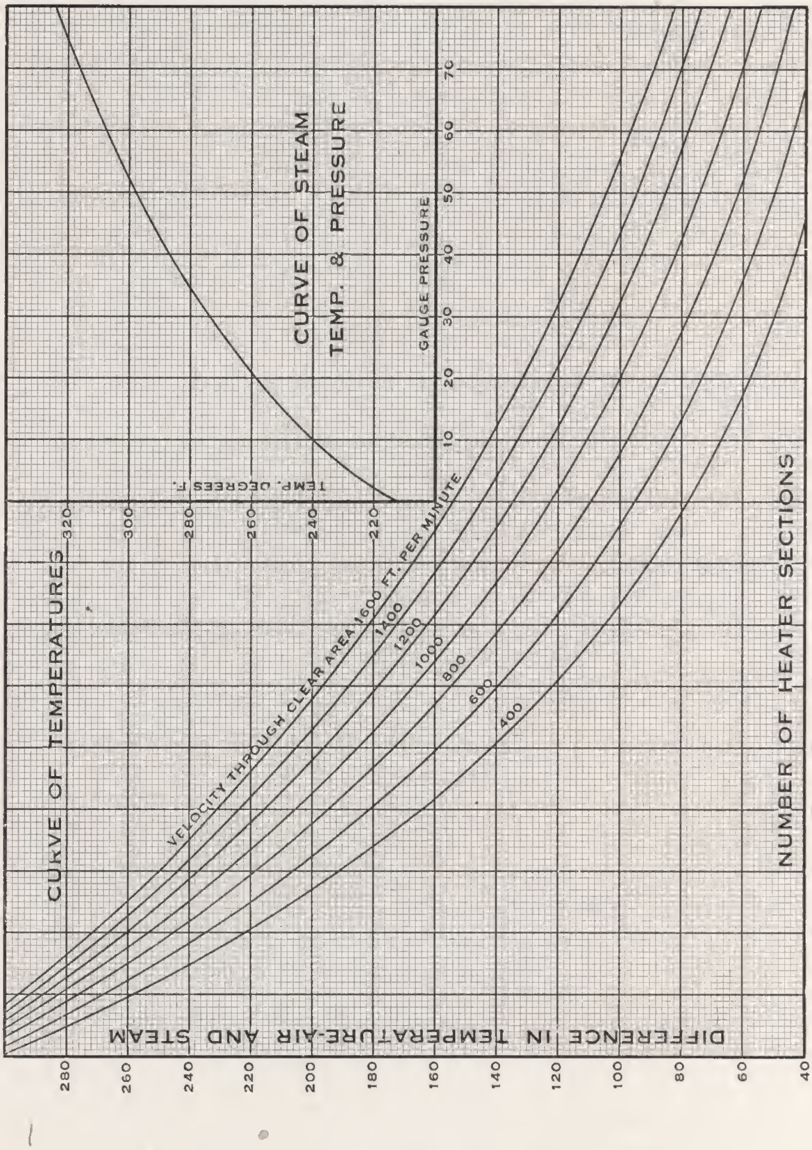
For the convenience of engineers who may desire results at other steam pressures and air temperatures than those given in table we include also the chart on page 136 which will be found very convenient and serviceable. The vertical scale gives the difference in temperature between steam and air, either initial or final as the case may be. The horizontal scale enables the depth of the heater to be determined, each division representing a section of four rows of pipe in depth. The depth of the heater is thus measured by the horizontal distance traveled in following the curve from the point of initial difference in temperature between steam and air to the point of final difference in temperature between steam and air. A separate curve is given for each velocity through the heater.

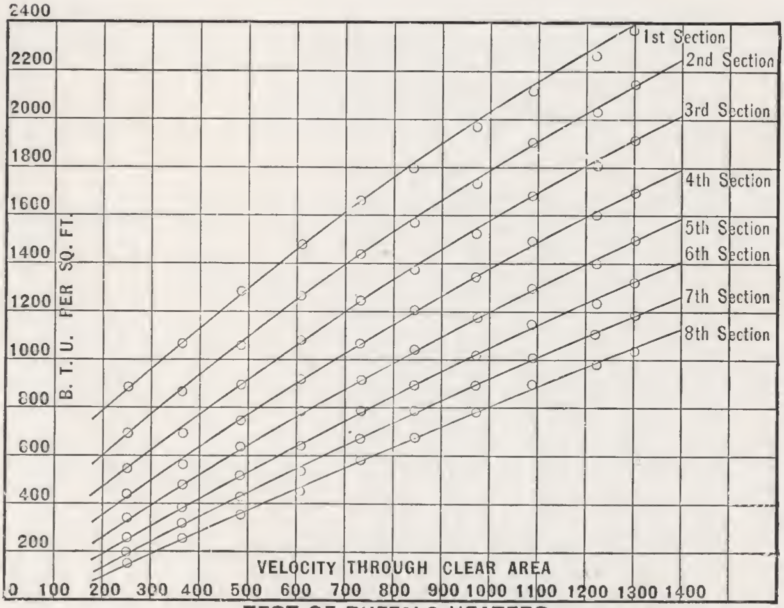
Example: Assume in a building that part of the air is returned at 60° F., and part taken from outdoors at 0° F. This makes entering air 30° F. Assume further that the air is to be heated to a final temperature of 146° F., and that the velocity through the clear area of the heater is 1000 feet per minute. Steam pressure 35 pounds. Refer to curves on page 136.

Temperature of steam at 35 pounds is 280° F. or there is 250° F. initial difference between steam and air. Final difference between steam and air is 134° F. We now find that by referring to the curve of 1000 feet velocity that there are six horizontal divisions between 250° F. and 134° F. In other words, six heater sections are to be used.

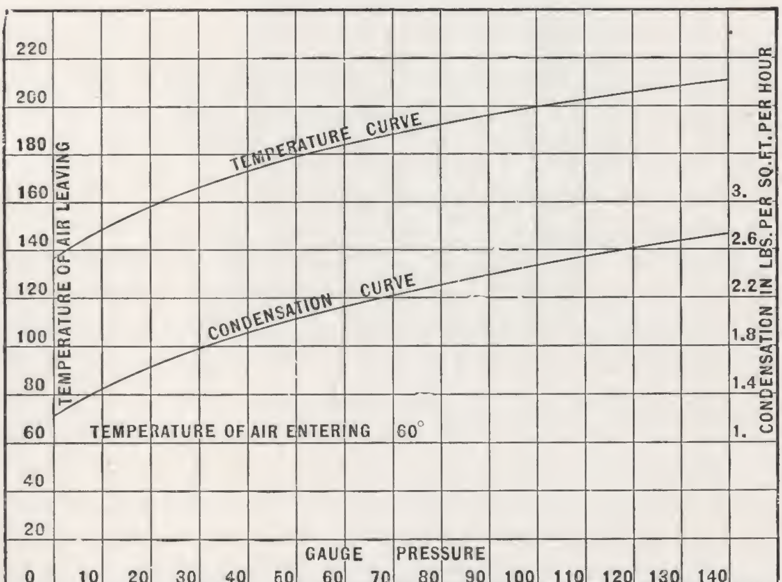
Rate of Condensation

The effect of air velocity and temperature upon the rate of condensation is shown very nicely by the graphical representation of an actual test, shown on page 137. It will be noted that the rate of transmission decreases with the increase in the temperature of the air in passing through successive sections of the heater but increases very rapidly with the increase in air velocity. The effect of steam pressure upon this rate of condensation and upon the temperature of the air is shown by the curve on bottom of page 137, for a heater having 24 rows of pipe in depth and having a velocity of air of 1000 feet per minute through the clear area and an entering temperature of air of 60°.


CURVE OF TEMPERATURES
CURVE OF STEAM
TEMP. & PRESSURE
GAUGE PRESSURE
10 20 30 40 50 60 70
TEMP. DEGREES F.
320 300 280 260 240 220
VELOCITY THROUGH CLEAR AREA 1600 FT. PER MINUTE
1400
1200
1000
800
600
400
NUMBER OF HEATER SECTIONS
1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17
DIFFERENCE IN TEMPERATURE-AIR AND STEAM
280 260 240 220 200 180 160 140 120 100 80 60 40



TEST OF BUFFALO HEATERS



EFFECT OF STEAM PRESSURE UPON TEMPERATURE AND CONDENSATION

TABLE OF AVERAGE INDOOR TEMPERATURES

Maintained at Various Outdoor Temperatures with
5 lbs. Steam Pressure

Outdoor Temp.	Average Indoor Temperatures							
−20	45.2	50.8	56.1	61.6	67.1	72.5	77.9	83.4
−15	48.9	54.3	59.7	64.9	70.3	75.6	80.9	87.3
−10	52.9	57.9	63.1	68.3	73.5	78.7	86.0	89.2
−5	56.3	61.4	66.5	71.6	76.8	81.9	87.0	92.1
0	60°	65°	70°	75°	80°	85°	90°	95°
5	63.7	68.6	73.5	78.4	83.2	88.1	93.0	97.9
10	67.4	72.1	76.9	81.7	86.5	91.3	96.0	100.8
15	71.0	75.7	80.3	85.1	89.7	94.4	99.1	103.7
20	74.7	79.3	83.9	88.4	92.9	97.5	102.1	106.6
25	78.4	82.9	87.3	91.8	96.2	100.7	105.1	109.5
30	82.1	86.4	90.8	94.1	99.4	103.8	108.1	112.4
35	85.8	90.0	94.3	97.5	102.6	106.9	111.2	115.3
40	89.4	93.6	97.7	101.8	105.9	110.0	114.2	118.2
45	93.1	97.1	101.2	105.4	109.1	113.2	117.2	121.1
50	96.8	100.7	104.7	108.5	112.4	116.3	120.2	124.0
55	100.5	104.3	108.1	111.9	115.6	119.4	123.3	126.9
60	104.2	107.8	111.6	115.2	118.8	122.6	126.3	129.8
65	107.8	111.4	115.0	118.6	122.1	125.7	129.3	132.7
70	111.5	115.0	118.5	121.9	125.3	128.8	132.4	135.6

Determination of Guarantees

The case often arises that a guarantee to heat a building to a certain specified temperature must be demonstrated at a much higher outside temperature than called for in the guarantee. It then becomes important to know the exact relation between increase in outside temperature and increase in inside temperature when apparatus is operated to its full capacity. This relation has been published for heating with direct radiation, but it varies considerably from the results obtained with the fan system. Naturally the rise in the indoor temperature will be less than the rise in outdoor temperature owing to the fact that the condensing capacity of the apparatus decreases with the temperature. With a fan system heater the condensing capacity has been shown to be directly proportional to the difference in temperature between steam and air, while with direct radiation it is not directly proportional owing to the variation in convection currents. The same relation between indoor and outdoor temperature may be shown to hold true whether the system was designed to take the air from outdoors entirely or to

recirculate air within the building. The formula expressing the relation between indoor and outdoor temperature in either case is,

$$T_r = \frac{T_r'(T_s - T_1) + T_s(T_1 - T_1')}{T_s - T_1'}$$

T_r=Temperature of building obtained with outside temperature T_1.

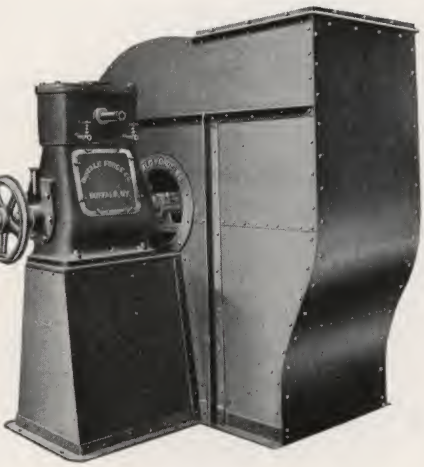
T_1=Any outside temperature at which test is made.

T_r'=Temperature of building guaranteed.

T_1'=Specified outside temperature.

T_s=Temperature of steam at pressure specified.

The table on page 138 gives corresponding indoor temperatures for various outdoor temperatures with guarantees at 60° to 95° in zero weather.



UP DISCHARGE PLANOIDAL FAN DIRECT CONNECTED TO DOUBLE VERTICAL, SINGLE-ACTING ENGINE

A TABLE OF AREA AND CIRCUMFERENCE OF CIRCLES

Diameter in Inches	Area		Circumference in Feet	One Side of a Square	Diameter in Inches	Area		Circumference in Feet	One Side of a Square	Diameter in Inches	Area		Circumference in Feet	One Side of a Square
	Square Inches	Square Feet				Square Inches	Square Feet				Square Inches	Square Feet		
1	.7854	.0054	.2618	.8862	35	962.1	6.681	9.163	31.0179	69	3739	25.97	18.06	61.1497
2	3.142	.0218	.5236	1.7724	36	1017.8	7.069	9.425	31.9042	70	3848	26.73	18.33	62.0359
3	7.070	.0491	.7854	2.6587	37	1075.2	7.467	9.686	32.7904	71	3959	27.49	18.59	62.9221
4	12.57	.0873	1.047	3.4549	38	1134.1	7.876	9.948	33.6766	72	4072	28.27	18.85	63.8083
5	19.63	.1364	1.309	4.4311	39	1194.5	8.296	10.21	34.5628	73	4185	29.07	19.11	64.9946
6	28.27	.1964	1.571	5.3174	40	1256.6	8.727	10.47	35.4491	74	4301	29.87	19.37	65.5808
7	38.48	.2673	1.833	6.2036	41	1320.2	9.168	10.73	36.3353	75	4418	30.68	19.63	66.4670
8	50.27	.3491	2.094	7.0898	42	1385.4	9.621	10.99	37.2215	76	4536	31.50	19.90	67.3500
9	63.62	.4418	2.356	7.9760	43	1452.2	10.08	11.26	38.1078	77	4657	32.34	20.16	68.4800
10	78.54	.5454	2.618	8.8623	44	1520.5	10.56	11.52	38.9444	78	4778	33.18	20.42	69.1500
11	95.03	.6600	2.880	9.7485	45	1590.4	11.04	11.78	39.8802	79	4902	34.04	20.68	70.0290
12	113.1	.7854	3.142	10.6347	46	1661.9	11.54	12.04	40.7664	80	5027	34.91	20.94	70.8950
13	132.7	.9218	3.403	11.5209	47	1734.9	12.05	12.30	41.6527	81	5153	35.78	21.21	71.8000
14	153.9	1.069	3.665	12.4072	48	1809.5	12.51	12.57	42.5839	82	5281	36.67	21.47	73.3500
15	176.7	1.227	3.927	13.2934	49	1885.7	13.09	12.83	43.4251	83	5411	37.57	21.73	73.5540
16	201.0	1.396	4.189	14.1796	50	1963.5	13.64	13.09	44.3113	84	5542	38.48	21.99	74.4460
17	226.9	1.576	4.451	15.0659	51	2043	14.19	13.35	45.9760	85	5675	39.41	22.25	75.4785
18	254.4	1.767	4.712	15.9521	52	2124	14.75	13.61	46.0838	86	5809	40.34	22.51	76.2170
19	283.5	1.969	4.974	16.8383	53	2206	15.32	13.88	46.9700	87	5945	41.28	22.78	77.1038
20	314.1	2.182	5.236	17.7245	54	2290	15.90	14.14	47.8562	88	6082	42.24	23.04	77.9871
21	346.3	2.405	5.498	18.6108	55	2376	16.50	14.40	48.7425	89	6221	43.20	23.30	78.8733
22	380.1	2.640	5.760	19.4970	56	2463	17.10	14.66	49.6287	90	6362	44.18	23.56	79.7621
23	415.4	2.885	6.021	20.3832	57	2552	17.72	14.92	50.5149	91	6504	45.17	23.82	80.6473
24	452.3	3.142	6.283	21.2694	58	2642	18.35	15.18	51.4012	92	6648	46.16	24.09	81.5389
25	490.8	3.409	6.545	22.1557	59	2734	18.99	15.45	52.2874	93	6793	47.17	24.35	82.4196
26	530.9	3.687	6.807	23.0419	60	2827	19.63	15.71	53.1736	94	6940	48.19	24.61	83.3060
27	572.5	3.976	7.069	23.9281	61	2922	20.29	15.97	54.0598	95	7088	49.22	24.87	84.1902
28	615.7	4.276	7.330	24.8144	62	3019	20.97	16.23	54.9061	96	7238	50.27	25.13	85.0760
29	660.5	4.587	7.592	25.7006	63	3117	21.65	16.49	55.8323	97	7390	51.32	25.39	85.9650
30	706.8	4.909	7.854	26.5868	64	3217	22.34	16.76	56.7185	98	7543	52.38	25.66	86.8500
31	754.7	5.241	8.116	27.4730	65	3318	23.04	17.02	57.6047	99	7698	53.46	25.92	87.7380
32	804.2	5.585	8.378	28.3594	66	3421	23.76	17.28	58.4910	100	7855	54.54	26.18	88.6280
33	855.3	5.940	8.639	29.2455	67	3526	24.48	17.54	59.3772					
34	907.9	6.305	8.901	30.1317	68	3632	25.22	17.80	60.2634					

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